

Proceedings of:

The Eighth Annual Meeting

The American Society for Precision Engineering

November 7-12, 1993

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Seattle, Washington

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a new focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology — and to represent all facets from research to application.

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These are exciting times for precision manufacturing. Scientific advances in materials, metrology, machine design and controls have extended the potential for design and manufacturing into the regime of nanotechnology. The nineties will bring an expanded, integrated view of precision manufacturing with increased emphasis on metrology and material forming processes. Driven by applications for mechanical parts (bearings and gears), electronic components (VLSI and electro-optics), materials (ceramics and electronic), adaptive control, and system integration, there will be a greater need for an improved understanding of metrology, machining, materials response, sensors, control, and machine design to achieve their goals. The aim of our Societies is to serve as a focus for the profession, and to emphasize the need for information exchange in this important area.

We are proud to present the program for the Eighth Annual Meeting of the ASPE, and pleased that the The Japan Society for Precision Engineering, JSPE, is a cooperating sponsor of this year's conference. This international forum for the exchange of ideas, for the discussion of solutions to problems, and for the presentation of results relating to precision engineering involves professionals in private industry, government laboratories and universities. The meeting offers the opportunity for multidisciplinary interaction and a view of the breadth of precision engineering.

The 1993 program reflects the vitality of our Societies, and the zest of this Annual Meeting.

See you in Seattle!

1993 ASPE Meeting Organizing Committee

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1993 Annual Meeting

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Preface

This book comprises the proceedings of the 1993 Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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Program

1993 ASPE Annual Meeting

Time	Sun., Nov. 7	Mon., Nov. 8	Tues., Nov. 9	Wed., Nov. 10	Thurs., Nov. 11	Fri., Nov. 12
8:00			Continental Breakfast			Technical Tours Boeing Everett Commercial Airplane Plant
9:00	Tutorial	Tutorial	Technical Session 1	Technical Session 4	Technical Session 6	
10:00			Break + Poster Session	Break	Break	
11:00				Technical Session 5	Technical Session 7	
12:00			Lunch + Committee Meetings	Lunch + Business Mtg	Lunch	
1:00						
2:00	Tutorial	Tutorial	Technical Session 2	OPEN (Exhibits Open)	Technical Session 8	
3:00			Break		Break	
4:00			Technical Session 3		Technical Session 9	
5:00						
6:00			Distinguished Lecture			
7:00		Welcome Reception	Dinner	Commercial Session/ Open Forum		
8:00			Hospitality Suite			
9:00				Hospitality Suite		
10:00						

Other Technical Sessions

Distinguished Lecturer

Tuesday, November 9, 6:00 pm

Norio Taniguchi

Norio Taniguchi has been a professor of Mechanical Engineering at Tokyo Science University since 1973. Prior to joining this University, he was employed at the Institute of Physical and Chemical Research in Tokyo and National Yamanashi University in Kofu, outside of Tokyo.

Dr. Taniguchi began his research on mechanism of machining processes on hard and brittle materials in 1940. The materials of interest included precious stones, quartz, silicon crystals, alumina crystals, ceramics, and optical glasses. Later, he directed his research toward non-conventional processing methods, and his initial interest was in ultrasonic machining of hard and brittle materials. Later still, he focused on various kinds of energy beam processing, using electric microwaves, electron beams, laser beams, etc. In a classic paper published in 1983, he organized the development of achievable machining accuracy.

His present research concerns ultraprecision processing using ion beam energy. Using ion beams, it is possible to fabricate ultraprecision and ultrafine products with nanometer accuracy and subnanometer resolutions. Dr. Taniguchi is also involved with unmanned manufacturing systems, especially automated assembly processes. He has taken on an important role in the progress of automated assembly technologies in Japan.

Dr. Taniguchi is an active member of the International Institution for Production Engineering Research (CIRP) and JSPE. He served as president of JSPE in 1970 and 1971.

Commercial Session and Open Forum

Wednesday, November 10, 7:00-9:00

Commercial Session Chair: William J. Bryan, 3M Company

Open Forum Chair: Robert J. Hocken, University of North Carolina - Charlotte

The commercial and open forum sessions will be held on Wednesday, November 10. In this special evening session, company representatives are being invited to make brief presentations on new products associated with precision engineering. You will have the opportunity to hear about limitations on performance of existing approaches, new approaches relevant to precision engineering and advances that can be expected in the future.

The open forum was created to encourage dissemination of new and significant results, as well as observations on precision engineering subjects. All interested conference participants can sign up at the Conference Registration Table for short, informal presentations.

Technical and Poster Paper Index

The technical program for the 1993 joint ASPE/JSPE Annual Meeting contains over 150 papers on current research from Asia, Australia, Europe, India, South America and the U.S. Authors from national laboratories, universities and industry will share their ideas and conclusions on a wide variety of topics.

Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster session. This year the poster papers will be close to both the exhibits and the technical sessions for the duration of the conference, and authors will be present to discuss their work on Tuesday, November 9 from 10:00 am to 12:00 pm.

In addition to a total of ten technical sessions, an Open Forum will be offered where recent results can be discussed, as well as a Commercial Session where manufacturers will give an update on their most recent innovations.

Session I Grinding

Tuesday, November 9, 1993, 8:30 - 10:00 am

Session Chair: Masakazu Miyashita (Ashikaga Institute of Technology)

1. **Influence of Wet Grinding Temperature on Surface Morphology of Fine Ceramics**
A. Hosokawa, H. Yasui (Kumamoto University), and K. Higuchi (Seiko Epson Corp.) page 13
2. **Ultraprecision Grinding of Optical Glasses for Obtaining Optical Surfaces**
Y. Namba, M. Abe (Chubu University) page 17
3. **Grinding With Directionally Aligned SiC Whisker Wheel**
K. Yamaguchi (Nagoya University), and I. Horaguchi (Toyota College of Technology) page 21
4. **Surface Characterization of Mirror-polished and Ductile-mode-ground Si Wafers**
N. Yasunaga (Tokai University), and Y. Samitsu, K. Abe, K. Hatta (Nippon Steel Co.) page 25

Session II Metrology I

Tuesday, November 9, 1993, 1:30 - 3:10 pm

Session Chair: Steven Patterson (UNC-Charlotte)

1. **A Full-aperture Optical Test for Large Convex Aspheric Mirrors**
J. H. Burge, D. S. Anderson (University of Arizona) page 29
2. **Near-Infrared Phase-modulation Interferometer for Surface Flatness Measurement**
P. de Groot, L. Deck (Zygo Corporation) page 33
3. **Advanced Applications of Scanning Probe Microscopy***
D. Grigg (Digital Instruments Inc.)
4. **Tunneling Current Characterization: An Application of an X-Ray Interferometric Calibrated Flexure Stage**
J. A. Miller (UNC-Charlotte), and S. T. Smith (University of Warwick) page 37
5. **3.39 μ m Infrared Multiwavelength He-Ne Laser Interferometer for Absolute Distance Measurement**
Y. Zheng, W. Ren, J. Liang (Tsinghua University) page 41

* Paper not available at press time.

Session III

Machine Design I

Tuesday, November 9, 1993, 3:30 - 5:10 pm

Session Chair: John Ziegert (University of Florida)

1. **Design Strategies for Ultraprecision, High Stiffness Feed System of Slide on Plain Bearing Guideways**
M. Daito (Nissin Machine Works, Ltd.), A. Kanai, M. Miyashita (Ashikaga Institute of Technology), and J. Yoshioka (NUTR K.K.) page 45
2. **Evaluation of a Tapered Roller Bearing Spindle for High-precision Hard Turning Applications**
M. A. Donmez, K. Harper, L. Greenspan (National Institute of Standards and Technology), and Y. Matsumoto, L. Keller (The Timken Co.) page 49
3. **Proposal of Multi-point Ultraprecision Slide Feed Mechanism by Making Use of Force-operated Actuator**
A. Kanai, M. Miyashita, S. Uchida (Ashikaga Institute of Technology) page 54
4. **Design Optimization of Herring Bone Air Bearings for Polygon Mirror Laser Scanners**
G. Lin, G. Huang, R. O. Warrington (Louisiana Tech University) page 58
5. **An Active Inherent Restrictor for an Infinite Stiffness Aerostatic Bearing**
H. Mizumoto, S. Arai (Tottori University), and Y. Kami (Nachi-Fujikoshi Co., Ltd.) page 62

Session IV

Metrology II

Wednesday, November 10, 1993, 8:00 - 10:00 am

Session Chair: Robert Hocken (UNC-Charlotte)

1. **Residual Error Compensation of a Vision-based Coordinate Measuring Machine**
M. Ananda (Pacific Precision Laboratories, Inc.), J. Raja (UNC-Charlotte), and T. D. Doiron, J. Zimmerman (National Institute of Standards and Technology) page 66
2. **Sampling Methods and Substitute Geometry Algorithms for Measuring Cylinders in Coordinate Measuring Machines**
U. Babu, J. Raja, R. J. Hocken (UNC-Charlotte) page 70
3. **Development of Noncontact-type Coordinate Measuring Machine — Measurement of Accuracy of Screw Thread Profile**
K. Maruyama, H. Tsuji, H. Kakimoto, M. Kurumisawa (Tokyo Institute of Technology) page 74
4. **SEM Linewidth Metrology for X-Ray Lithography Masks**
M. T. Postek, J. R. Lowney, A. E. Viadar, W. J. Keery, E. Marx, R. D. Larrabee (National Institute of Standards and Technology) page 78
5. **Two New Probes for a Coordinate Measuring Machine**
J. A. Stone, Jr., L. Carroll, G. Caskey, S. Phillips (National Institute of Standards and Technology), R. Resnick, J. Rose (Extrude Hone), and K. Lau (Automated Precision, Inc.) page 82
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J. C. Ziegert, C. D. Mize (University of Florida) page 86

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Session Chair: Don Lucca (Oklahoma State University)

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D. Engelhaupt (University of Alabama/Huntsville), and R. W. Rood, C. Griffith (NASA Marshall S. F. C.) page 90
2. **Production of X-Ray Optics by Diamond Turning and Replication Techniques**
S. C. Fawcett (NASA Marshall S. F. C.), and D. E. Engelhaupt (University of Alabama/Huntsville) page 95
3. **Design and Manufacture of Slit Patterns for Gamma Ray Imaging**
P. V. Pistecky, H. F. van Beek, H. van der Wulp (Delft University of Technology) page 99
4. **Prospects for X-Ray Optics From Atomic Layer Growth**
J. M. Thorne, J. K. Shurtleff, D. D. Allred, J. D. Phillips (Brigham Young University) page 103

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Session Chair: Steven Fawcett (NASA/Marshall S.F.C.)

1. Precision Ion Figuring System Development

T. W. Drueding, T. G. Bifano (Boston University), and S. C. Fawcett (NASA Marshall S. F. C.) page 107

2. Nanoscale Plasticity in Silica Glass

J. N. Glosli, D. B. Boercker, A. Tesar, J. Belak (Lawrence Livermore National Lab.) page 111

3. Ion Beam Modification of Silicon Carbide for Improved Machinability

W. K. Kahl, C. M. Egert, R. D. Seals (Oak Ridge National Lab.), and C. J. McHargue (University of Tennessee) page 115

4. Ion Beam Sharpening of Quadrangular and Triangular Pyramidal Diamond Indenters or Probes

I. Miyamoto, S. Naito, S. Konta (Science University of Tokyo) page 118

5. Development of Optical Fiber Connector Polishing Machines and Ellipsometric Characterization of the Polished Surface Damage

F. Ohira, S. Matsui, K. Matsunaga, T. Saitoh, Y. Kikuya (NTT Interdisciplinary Research Labs) page 122

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Session Chair: Anthony Gee (Cranfield University)

1. Precision Machining and Measurement Organized by Miniature Robots

H. Aoyama, A. Sasaki, T. Kubo (Shizuoka University) page 126

2. Application of Advanced Ceramics Materials to Sliding Guideways (Friction and Wear Characteristics Under Fluid and Solid Lubrication)

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H. Hachiuma, M. Nakao, Y. Hatamura (The University of Tokyo) page 142

2. Characterization of Surface Particulate Contamination on Liquid Crystal Optics for Laser Fusion*

L. D. Lund, A. Rigatti (University of Rochester), and P. Glenn, J. Glenn (Bauer Associates)

3. On Replication and 3D Stylus Profilometry Techniques for Measurement of Plateau-honed Cylinder Liner Surfaces

R. Ohlsson, B. Rosén (Chalmers University of Technology) page 146

4. Effect of Tool Wear on Surface Finish, Texture, and Tool Forces

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* Paper not available at press time.

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R. C. Montesanti, P. J. Davis (Lawrence Livermore National Lab.) page 158
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M. Tanaka (Nippon Seiko K.K.) page 166
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M. Walter (Rank Taylor Hobson Inc.) page 205
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Technical Papers



A S P E

Influence of Wet Grinding Temperature on Surface Morphology of Fine Ceramics

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1. Introduction

The hard-brittle materials such as fine ceramics are finished mainly by grinding with diamond wheels due to their mechanical properties of high hardness and brittleness. Several investigations have been made on the grinding of fine ceramics. Only a few attempts, however, have been done on the grinding temperature because of the limitation of the thermometry of an insulation medium as ceramics.

In this study, the wet grinding temperatures of fine ceramics are measured by means of an infrared radiation pyrometry in which a PbS-cell is mounted just below the workpiece and the influence of grinding temperature on the finished surface morphology is examined. In addition, the grinding heat partitions into workpiece are calculated based on the simple model of heat transfer in wet grinding and the influence of mechanical properties of workpieces and the grinding parameters on the abrasive-workpiece interaction are investigated.

2. Experimental Procedure

The surface grinding of plunge up-cut type was carried out at a constant wheel depth of cut as shown in Fig.1. Two types of engineering ceramics — normally sintered silicon carbide (SSC) and normally sintered silicon nitride (SSN)— are chosen as workpieces. The Grinding fluids are a water-based soluble type (1:50 in water) and a straight oil, and are applied from a nozzle to the grinding zone as illustrated in Fig.1.

The grinding temperature is measured by

means of an infrared radiation pyrometry. A photoconductive PbS-cell is mounted just below the workpiece and accepts the infrared rays radiated from the bottom of a small hole which extends nearly to the workpiece surface. This direct-type pyrometer makes it possible to measure the temperature above approximately 50°C [1].

The experimental conditions and the characteristics of workpiece materials are summarized in Table 1 and Table 2, respectively. Prior to each experiment, the topography of the wheel surface is adjusted by truing and/or dressing listed in Table 1, so that the average depth of chip pocket is controlled in the rage of 25–35 μm .

3. Grinding Zone Temperature and Heat Partition into Workpiece

The pyrometer measures the temperature histories for various depths below the ground surface. Therefore, the peak grinding zone temperature θ_m is estimated by extrapolating the peak temperature distribution in a workpiece.

Based on the report that the temperatures in the vicinity of cutting grains are high even

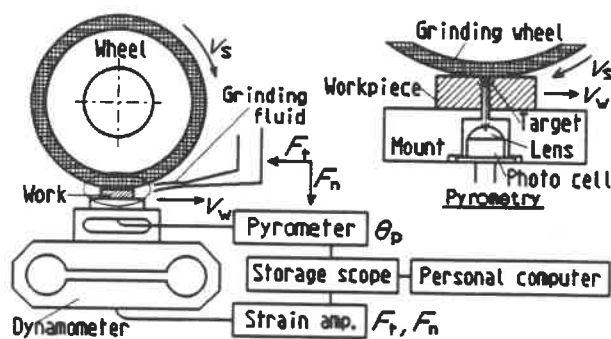


Fig.1 Experimental arrangement

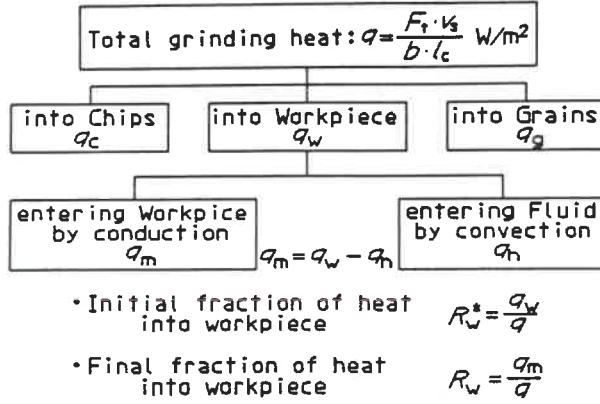


Fig.2 A simplified model of heat transfer in wet grinding without film boiling of grinding fluid in wheel-work interference zone

Table 1 Experimental conditions

Grinding wheel : SDC140N75B $d_s=200$ mm, $b_s=10$ mm	•Truing (SD100Q75M) $v_w=2$ m/min $s=50$ μ ■/rev $a=5$ μ ■
Wheel speed : $v_s=25$ m/s Work speed : $v_w=0.2-15$ m/min Depth of cut : $a=2-30$ μ ■	Soluble(1:50)
Grinding fluids : Water-based soluble(1:50) Straight oil Flux : 10 L/min	•Dressing (WA600H7V & #600) $v_w=2$ m/min $a=50$ μ ■/pass Soluble(1:5)

Table 2 Characteristics of workpiece materials

Workpiece material		S S C	S S N
Density : ρ_w	g/cm ³	3.15	3.20
Specific heat : c_w	J/(kg·K)	650	712
Thermal conductivity : k_w	W/(m·K)	100	16.7
Fracture toughness : K_{Ic}	MN/m ^{3/2}	4.0	5.0
Hardness :	HV	2500	1600

if the film boiling of grinding fluid does not occur [3], the following simple model of heat transfer in wet grinding may be introduced as shown in Fig.2. The total grinding energy(=q) is separated into grains, chips and workpiece(=q_w) as heat during contacts between grains and workpiece (heat into grinding fluid is negligible) and part of q_w(=q_f) flows into grinding fluid by convection heat transfer through the entire workpiece surface. Thus, two kinds of energy partitions are described: the initial fraction $R_w^*=q_w/q$ and the final one $R_w=(q_w-q_f)/q$.

In calculating R_w^* and R_w , the moving heat source theory with convecting cooling

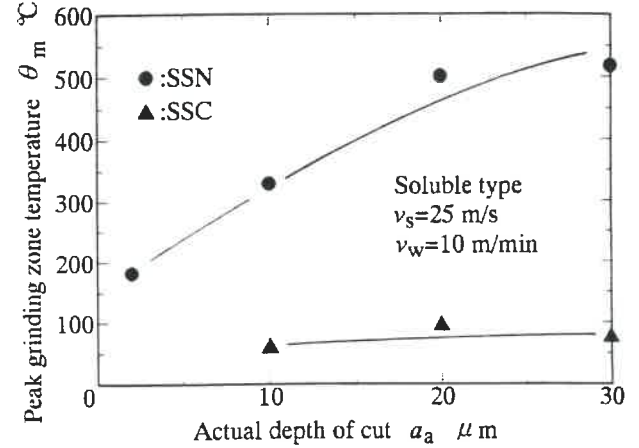


Fig.3 Variation of peak grinding zone temperature with actual wheel depth of cut for two kinds of fine ceramics

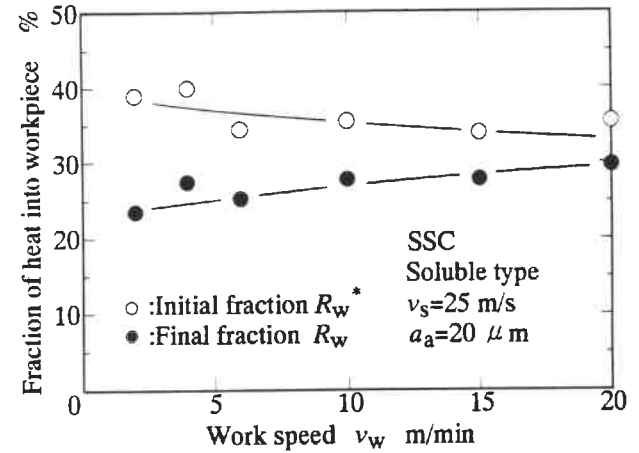


Fig.4 Relationship between work speed and fractions of heat into workpiece for SSC

($h=40$ kW/(m²·K) with water-based fluid without film boiling) is applied [2].

4. Results and Discussion

4.1 Peak Grinding Zone Temperature and Heat Partition

Figure 3 shows the relationship between the peak grinding zone temperature θ_m and the actual wheel depth of cut a_a for two kinds of fine ceramics. The value of θ_m of SSC is kept roughly constant below 100°C with water-based grinding fluid although θ_m of SSN rises considerably to 500°C or higher with a_a .

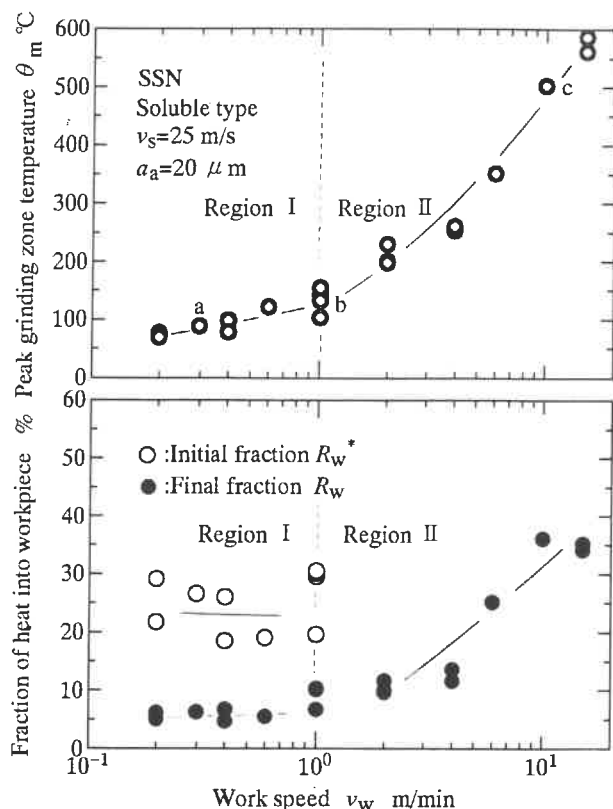
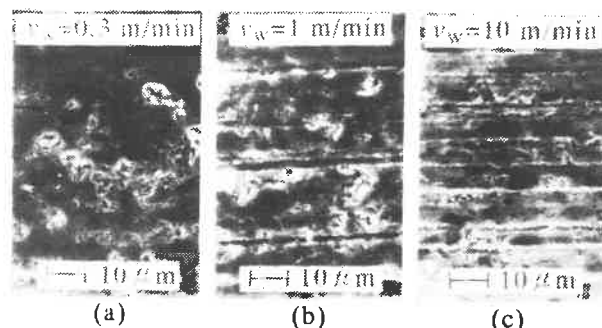


Fig.5 Influences of work speed on peak grinding zone temperature and fractions of heat into workpiece for SSN with water-based soluble fluid

Figure 4 shows the relationship between two kinds of heat partitions and work speed v_w . The initial fraction R_w^* decreases slightly from approximately 40% to 35% and the final fraction R_w increases from about 25% to 30% as work speed increases.

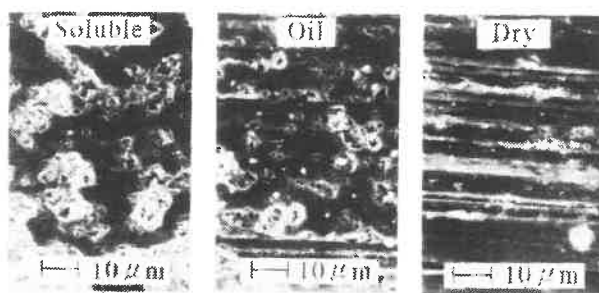
Judging from Figs.3 and 4, it seems reasonable to suppose that the abrasive-workpiece interaction of SSC in wet grinding with soluble fluid is not affected very much by depth of cut and work speed.

On the other hand, the variations of θ_m and heat partitions with work speed for SSN are shown in Fig.5. Here the surface temperature is seen to generally rise as work speed increases, but two regions are observed: Region I — small values of v_w (<1 m/min); Region II — large values of v_w (>1 m/min). There is a slight increase of θ_m with v_w in region I and a rapid rise in region II, and the similar relationship are



SSN, Soluble, $v_s=25$ m/s, $a_a=20$ μ m

Fig.6 SEM microphotographs of the finished surfaces denoted by "a", "b" and "c" in Fig.5



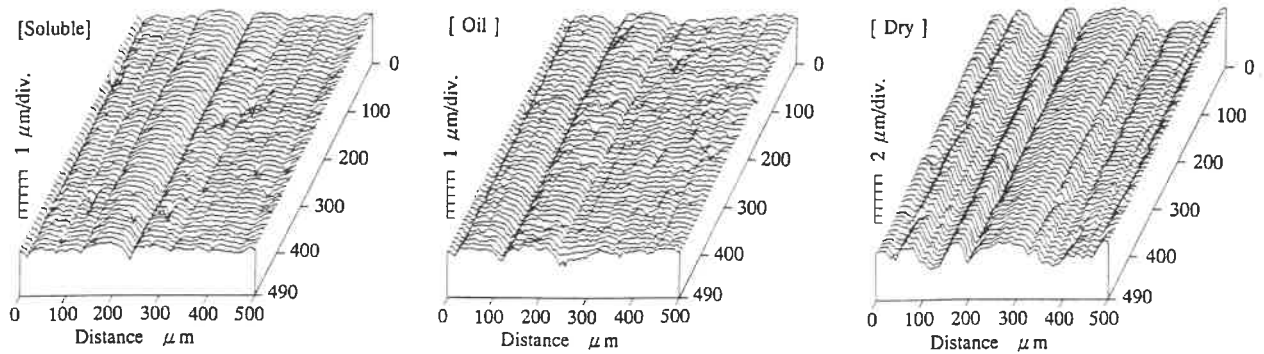
SSN, $v_s=25$ m/s, $v_w=0.2$ m/min, $a_a=20$ μ m

Fig.8 Surface roughness profiles of the finished surfaces corresponding to those in Fig.7

obtained for the variation of R_w versus v_w . It can be estimated from the figure that the abrasive-workpiece interaction is changed from region I to II as the grinding temperature rises.

4.2 Finished Surface Morphology

Figure 6 shows the SEM microphotographs of the finished surfaces denoted as "a", "b" and "c" in Fig.5. It can be seen from these photographs that the surface texture of "a" is highly fragmented with less flow like SSC. The clear grinding grooves, on the contrary, are observed in surface-"c" in which plastic flow area is dominant. The surface-"b" seems to be intermediate between "a" and "c". The main reasons for changing in surface morphology are estimated as follows: 1) Change in chip geometry with v_w . 2) Change in mechanical properties of workpiece with temperature.



SSN, $v_s=25$ m/s, $v_w=0.2$ m/min, $a_a=20$ μ m

Fig.8 Surface roughness profiles of the finished surfaces corresponding to those in Fig.7

Figures 7 and 8 show the SEM micro-photographs and surface roughness profiles of the finished surface of SSN, respectively, ground under the same grinding conditions but grinding fluid. It is clear that more flow with less fragmentation of the finished surface is observed in dry grinding and more fragmentation with less flow in wet grinding with soluble fluid ($\theta_m \approx 70^\circ\text{C}$). Accordingly the flow area increases as the grinding zone temperature rises under almost same chip geometry. In addition, several pits, which may be caused by brittle fracture, are appeared only in wet grinding as shown in Fig.8. Judging from the above, it may be possible to say that the abrasive-workpiece interactive is affected by grinding zone temperature in the case of SSN so that the surface morphology and the heat partition into workpiece are changed considerably. There is, however, room for further investigation on this problem.

5. Summary

Influences of grinding conditions on the grinding temperature of SSN are greater than that of SSC because of their dependence of mechanical properties on temperature. The maximum grinding zone temperature of SSC is kept below 120°C with water-based soluble grinding fluid under conventional grinding conditions, in which the finished

surface is highly fragmented with less flow.

In the case of SSN, two types of surface morphologies are observed corresponding to the grinding temperature in the work speed range of 0.2–15m/min. Highly fragmented ground surface like SSC is observed below the maximum grinding zone temperature of $100\text{--}200^\circ\text{C}$ and plastic flow area increases as the grinding zone temperature rises.

While the fraction of heat into workpiece of SSC does not vary very much with both work speed and wheel depth of cut, it increases with work speed above 1m/min in the case of SSN.

The abrasive-workpiece interaction of fine ceramics, especially SSN may be affected by the grinding zone temperature, so that the surface morphology varies with grinding conditions.

Acknowledgments

The authors are indebted to Noritake Diamond Industries Co., Ltd. and Yushiro Chemical Industry Co., Ltd. for providing the wheels and the grinding fluids, respectively.

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ULTRA-PRECISION GRINDING OF OPTICAL GLASSES FOR OBTAINING OPTICAL SURFACES

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1. Introduction

There arises a big demand for smooth glass surfaces not only for conventional optical use but also for electronic use such as glass plates for liquid crystal displays and glass discs for magnetic hard discs. Optical glasses have been finished by a traditional pitch polishing which is a labor intensive and duty work, though small aspherical optics have been recently produced by a precision molding process. In order to solve the above-mentioned demand, new ultra-precision machining process having higher productivity is expected to develop. In recent years, the ductile mode grinding of glass has been developed and expected to reduce the polishing time.

This paper deals with a new technology for making optical surfaces on glasses using the ultra-precision grinding without any polishing process.

2. Experiments

Various kinds of optical glasses such as FCD1, BSC7, BaCD5, BaCD16, LaC12, LaC14, F3, FD6, NbF1, TaF1 and LHG8 of HOYA Corp. were sawed and lapped into $18 \times 18 \times 8$ mm in dimensions. The ultra-precision surface grinder [1] having a glass-ceramic spindle of extremely low thermal expansion and various kinds of cup-typed resinoid-bonded diamond wheels as shown in Table 1 were used in this study. Grinding force has been measured with a dynamometer of 9257A of Kistler Corp. and the ground surface roughness has been measured with various methods, such as stylus, optical and AFM.

Table 1 Grain size of used diamond wheels.

Diamond wheel	Average grain size	Grain size
SD3000-75-B	4 (μm)	2- 6 (μm)
SD2000-75-B	8.5	5- 12
SD1500-75-B	11.5	8- 15
SD1200-75-B	14	8- 20
SD1000-75-B	18.5	12- 25
SD 800-75-B	30	20- 40
SD 400-75-B	50	40- 60
SD 200-75-B	100	90-110

3. Results and discussion

A variety of optical glass samples were ground with different diamond wheels under various grinding conditions. From a Nomarski microscopic observation, the ground surfaces can be classified into 3 modes, that is, fracture mode, ductile & fracture mode and ductile mode [2]. It is clarified by the fluoric acid etching of ground glass that the ground glass surface in the ductile mode grinding does not contain any micro-crack under the finished surface [2]. The grinding mode strongly depends upon the grain size of diamond wheels, and Fig. 1 shows the relationship between the ground surface roughness on NbF1 glass and average grain size of used diamond wheels. It is clear that the surface roughness depends upon the average grain size, and smoother surface can be obtained by using a diamond wheel of finer grain. There is a gap on the curve in Fig. 1 and the gap shows the boundary between the ductile mode grinding and ductile & fracture mode grinding. A ground surface with SD1200-75-B wheel shows the ductile mode grinding

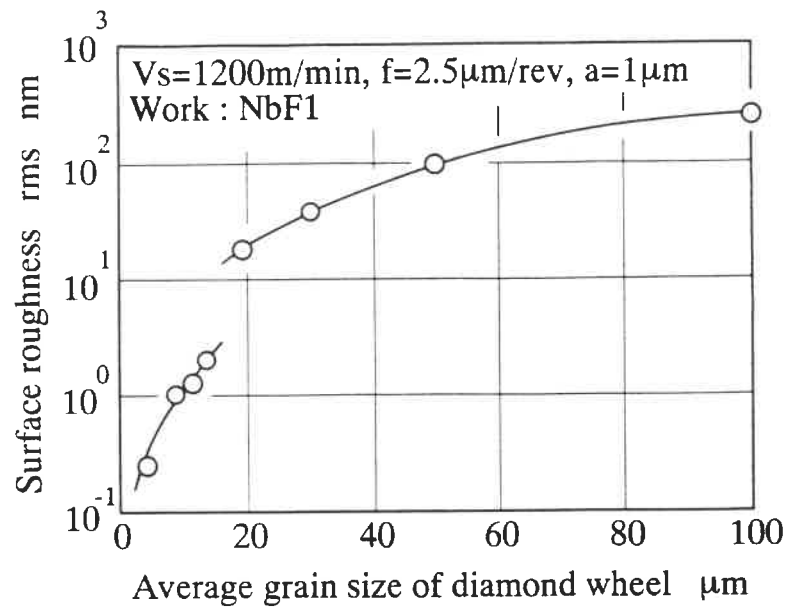


Fig. 1 The relationship between the ground surface roughness and average grain sizes of grinding wheels.

and the range of diamond grain size is 8-20μm, and a ground surface with a SD1000-75-B wheel shows the ductile & fracture mode grinding. So, it is concluded that NbF1 glass can be ground in the ductile mode with the grinding wheels having diamond grains less than 20μm in grain size.

Figure 2 shows the relationship between the surface roughness on ground NbF1 glass and feed per wheel revolution. The roughness strongly depends upon the feed per wheel revolution, and a smaller feed can perform a smoother surface. From the results of grinding experiments on

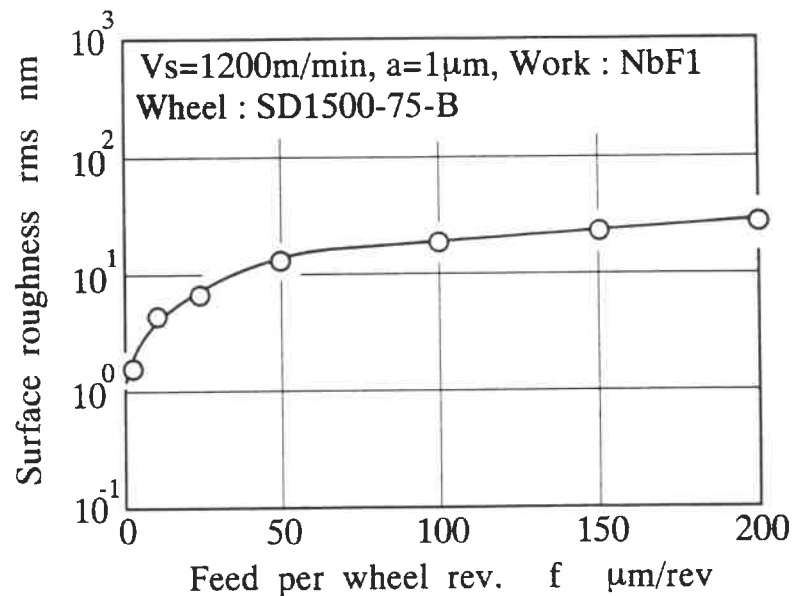


Fig. 2 The relationship between the ground surface roughness and feed per wheel revolution.

NbF1 glass samples under the conditions of depth of cut $a=0.2-10\mu\text{m}$, wheel peripheral speed $V_s=400-1200\text{m/min}$, it is clarified that the ground surface roughness does not depend upon the depth of cut and wheel peripheral speed in the ductile mode grinding.

From the surface roughness measurements with various methods such as stylus, optical and AFM, it is clarified that super-smooth surfaces (less than 0.2nm rms) better than optically-polished ones and the flatness of $\lambda/5$ can be obtained by the ultra-precision one path grinding without any polishing process. Higher productivity will be obtained by increasing the wheel peripheral speed and feed simultaneously.

It is easy to perform the ductile mode grinding using finer diamond grains, however, the grinding force suddenly increases with decrease of grain size as shown in Fig. 3. Higher grinding force may worsen the accuracy in forms and dimensions of machined parts. The grinding force in the ductile mode is much larger than that in the fracture mode which introduces very rough surfaces. The normal grinding force is the function of grain size of a wheel, feed per wheel revolution, depth of cut, material to be machined, grinding area and grinding time.

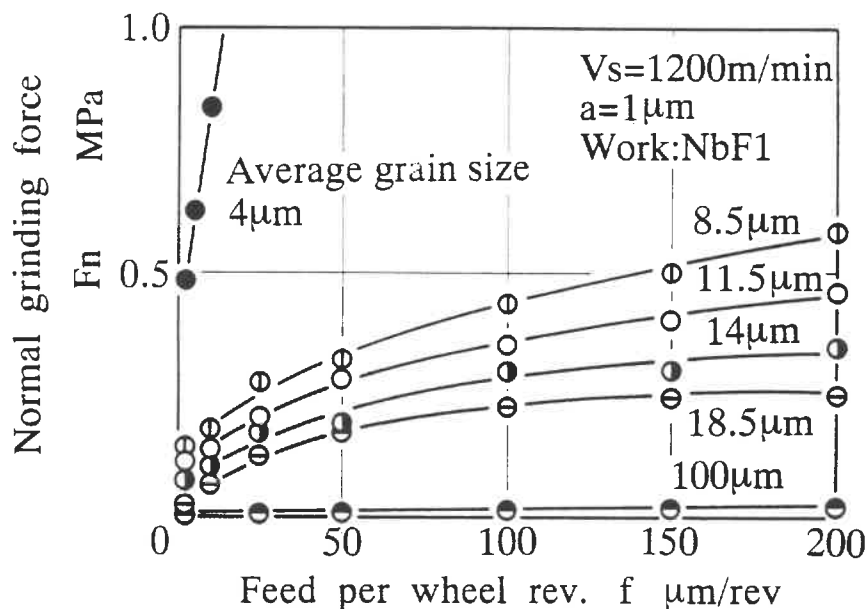


Fig. 3 The relationship between the grinding force and average grain sizes of used grinding wheels.

In the case of conventional lapping or polishing process, the finished surface roughness is the function of the hardness of material to be machined, however, the surface roughness does not a function of the material hardness in the ultra-precision grinding. The grinding force and surface roughness are not affected by the hardness of glasses. Figure 4 shows the relationship between the ground surface roughness and normal grinding force as a parameter of various optical glasses. It is clarified that there is a clear relationship between the ground surface roughness and normal grinding force if a variety of optical glass samples are ground under the same grinding conditions. The ground surface roughness decreases with the increase of grinding force. There may be a key to solve the mechanism of ductile mode grinding on brittle glasses.

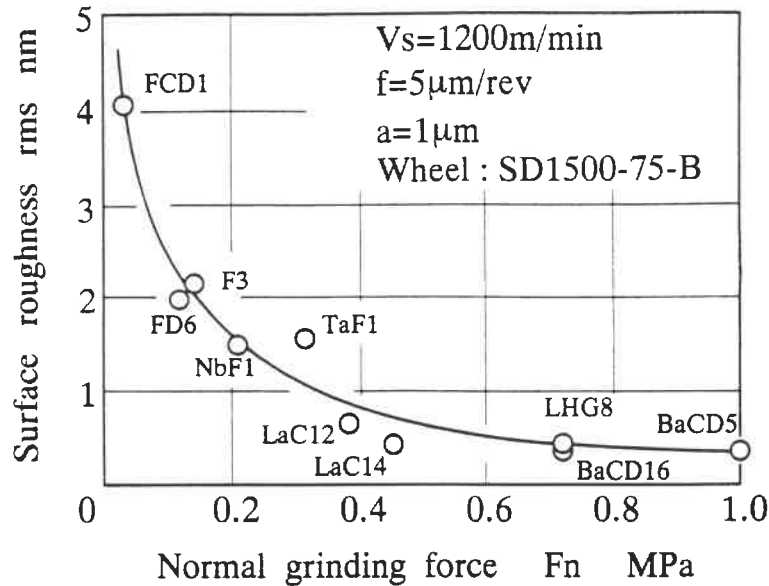


Fig. 4 The relationship between the ground surface roughness and normal grinding force on various optical glasses.

4. Conclusions

A variety of optical glasses have been ground by the ultra-precision surface grinder with various diamond wheels in mesh size under machining conditions of a wide range, and the following conclusions may be made on the results of this study.

1. Ductile mode grinding on NbF1 glass can be performed by using a resinoid-bonded diamond wheel of less than $20 \mu\text{m}$ in grain size.
2. The ground surface roughness is a function of grain size of diamond wheel, feed per wheel revolution, and not a function of the wheel depth of cut, wheel peripheral speed and hardness of glass. The surface roughness is a function of the grinding force.
3. Super-smooth surfaces better than optically-polished surfaces and the flatness of $\lambda/5$ have been obtained by the ultra-precision grinding without any polishing process.
4. The grinding force strongly depends upon the grain size of a diamond wheel, kind of glass and feed per wheel revolution.

Acknowledgments

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GRINDING WITH DIRECTIONALLY ALIGNED SiC WHISKER GRINDING WHEEL

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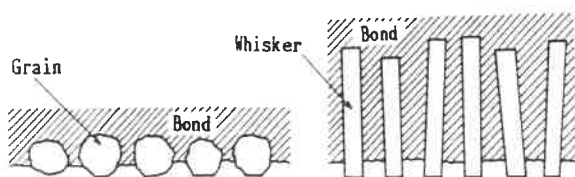
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1. INTRODUCTION

A grinding wheel essentially consists of a large number of abrasive grains, held together by a suitable bond. The most commonly used abrasives are grains of aluminum oxide (Al_2O_3), silicon carbide (SiC) and diamond. The surface finish requirement for grinding wheels is mainly determined by the grain size and strength of the bond. The bonding area for fine grains is small and thus the grains may be torn from the bond even under normal working conditions. Whiskers (fine fibers) on the other hand, have a larger bonding area than abrasive grains. Besides this, the diameter of the whisker is in the range of few micrometers. When the whiskers are set on end normal to the grinding surface, the fine end of the whiskers become the cutting edges. Thus, a high surface finish can be expected when using whiskers as the cutting edges of a grinding wheel. This paper deals with the procedure for making a grinding wheel with whiskers that are directionally aligned and normal to the grinding surface.

2. THEORETICAL CONSIDERATIONS OF WHISKER ALIGNMENT PROCESS

Figure 1 shows models of grinding wheel. The directionally aligned whiskers set normal to the grinding



(a) Abrasive grains (b) SiC whiskers
Fig.1 Model of the grinding wheel

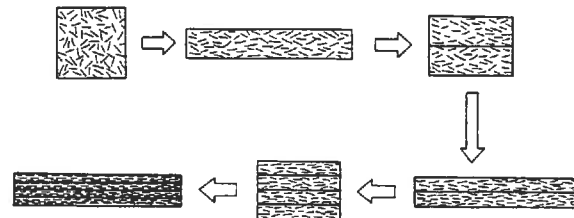


Fig.2 Procedure of directional alignment

surface as shown in 1(b) are characteristically fixed more securely than the abrasive grains that are shown in 1(a). Figure 2 shows the procedure carried out to align the whiskers. In this process, a mixture of SiC whiskers and liquid phenolformaldehyde resin is rolled into a cylindrical mass. This is then folded and rolled again. The process of repeated rolling and folding of the cylindrical mass produces directionally aligned whiskers.

When the cylindrical mass is elongated n times, its diameter is reduced by a factor of $n^{-1/2}$ and the whiskers are aligned in the direction of elongation. Suppose the angle changes from θ to ϕ , where θ and ϕ are projected angles depicted in two dimensional planes corresponding to before and after elongation, respectively, as shown in Figure 3. If we consider an elemental mixture that includes a whisker, the inclination angle ϕ can be given by $\tan\phi = n^{-3/2} \tan\theta$.

The initial state is assumed to have uniformly distributed whiskers in all directions and hence the probability density $P(\theta)$ is a constant. From the relation $\int_0^{\pi/2} P(\theta) d\theta = 1$ and $P(\phi) = P(\theta) d\theta / d\phi$ the probability density of the elongated mass, $P(\phi)$, can be obtained as follows:

$$P(\phi) = \frac{2}{\pi} \cdot \frac{1}{n^{3/2} \sin^2 \phi + n^{-3/2} \cos^2 \phi} \quad (1)$$

The calculated values of equation (1) are graphically represented in Figure 4. As shown in the figure, when the cylindrical mass is rolled and has been substantially elongated to 100 times

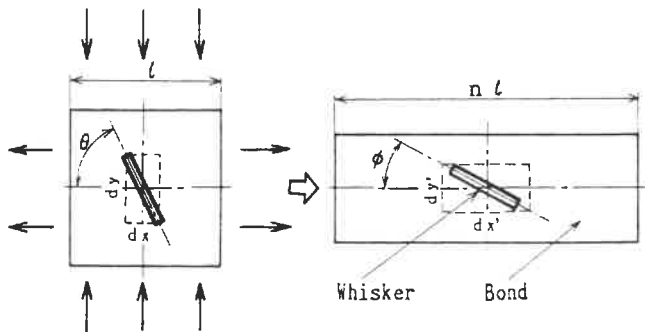


Fig.3 Variation of inclination angle under the process of directional alignment

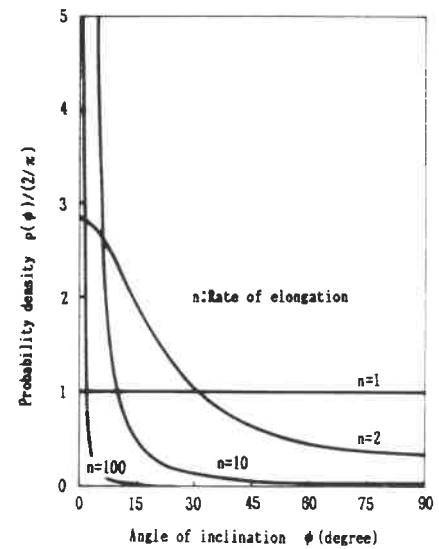


Fig.4 Azimuth distribution of whiskers

the length, the whiskers are aligned to within a few degrees of the direction of elongation.

3. CONSTRUCTION OF THE GRINDING WHEEL

Equal weight mixtures of SiC whiskers and liquid phenolformaldehyde resin are mixed thoroughly. This mixture is then rolled and elongated into a two fold length. It is then cut in half and over lapped. This is rolled into a two fold length again. The repetition of this process results in directionally aligned whiskers as shown in Figure 5. After completing the process of alignment the mixture is solidified and hardened in a furnace and finally formed into wheel segments. Figure 6 shows the schematic diagram of a straight cup type grinding wheel. The grinding operation was carried out using NC vertical milling machine at a low speed with water soluble grinding fluid. The work material was hardened die steel SKD 11 (AISI D2) and its hardness is HRC60.

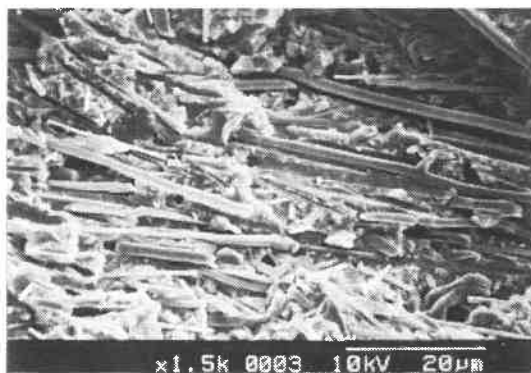


Fig.5 SEM micrograph of the cross-section of the grinding wheel

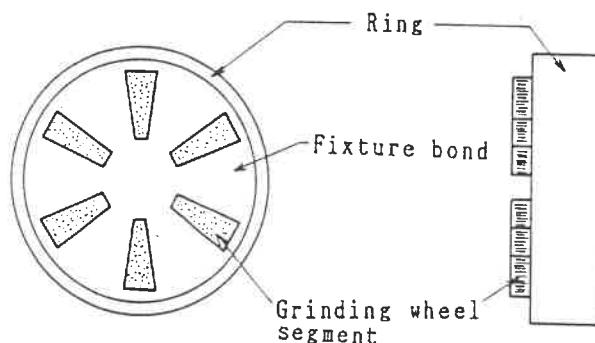


Fig.6 Schematic diagram of the grinding wheel

4. CHARACTERISTICS OF THE FINISHED SURFACE

Figure 7 shows the roughness curves of the finished surfaces of the hardened die steel, silicon wafer and glass were measured by a needle type roughness meter and Figure 8 shows the surface roughness of the die steel measured by Phase Measuring Interferometer. The R_{max} and R_a surface roughness values are $0.04 \mu m$ and $0.004 \mu m$, respectively. Figure 9 shows the ground surface has a mirror-like finish. Figure 10 and 11 show SEM micrographs of the grinding wheel surface and the ground surface, respectively.

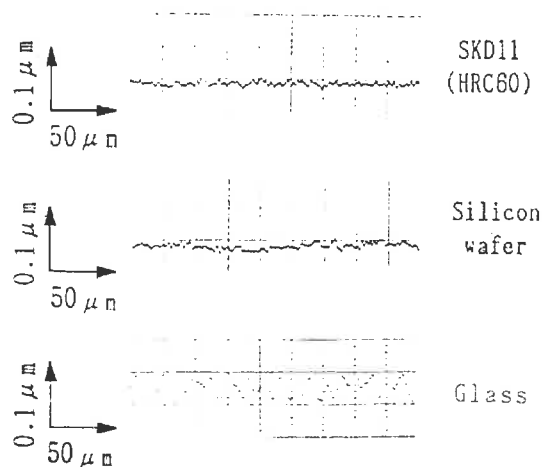


Fig.7 Roughness curves of various ground surfaces

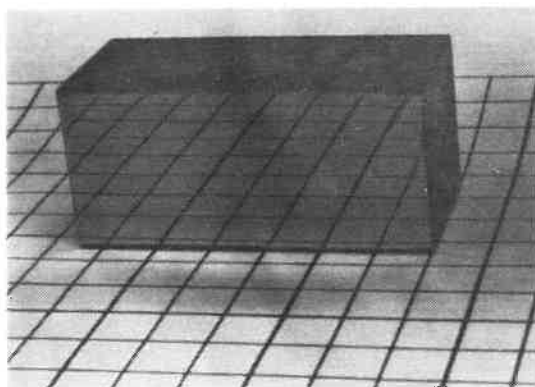


Fig.9 Photograph of the ground surface

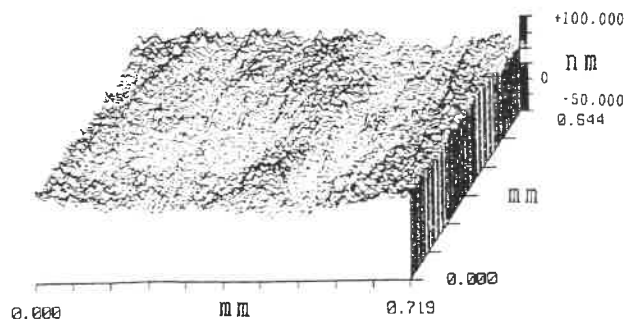


Fig.8 Surface roughness of SKD11 measured by Phase Measuring Interferometer

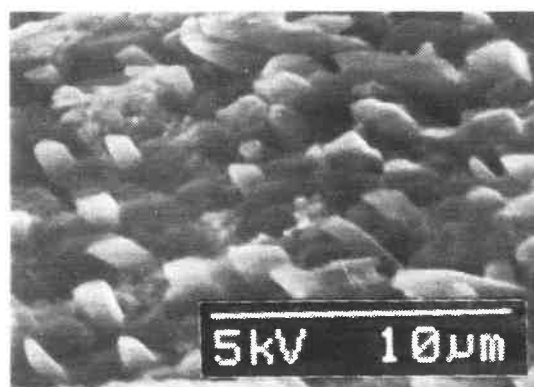


Fig.10 SEM micrograph of the grinding wheel surface

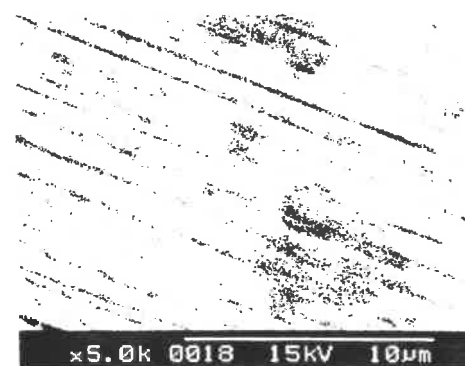


Fig.11 SEM micrograph of the ground surface

The above mentioned results indicate that a high surface finish can be obtained by using SiC whiskers as the cutting edge of a grinding wheel. For practical application, however, a study of a wide range of manufacturing conditions such as the mixing ratio of whiskers, the kind of bond, and grinding conditions etc. must be done in detail.

Surface Characterization of Mirror-polished and Ductile-mode-ground Si Wafers

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1 Features of current Si wafer polishing process

Higher flatness and smoothness are required for high quality Si wafers which will be used in coming highly integrated devices. Table I shows predictive values required for Si wafer production for DRAM. For example, the flatness value of LTV (Local Thickness Variation) should be less than $0.3 \mu\text{m}/25\text{mm}^2$ and $0.2 \mu\text{m}/30\text{mm}^2$, to 64MDRAM and 256MDRAM, respectively. Furthermore, microroughness of the wafer surfaces are desired to be much smaller for obtaining higher dielectric breakdown field intensity of thin-oxide films on wafer surfaces.

Almost all commercial Si wafers are today produced through conventional lapping and polishing processes. In the polishing process, which is the most influential in determining the final shape accuracy and the microroughness of the Si wafers, cloth-like, soft resin pads which generally deform rheologically are used as polishers for the primary polishing. It has already been found that the total wafer-size flatness represented by NTV (Nonlinear Thickness Variation) was more improved as the instantaneous elastic deformation of the pad, which is theoretically proportional to the reciprocals of the coefficient of elasticity E_1^* , was smaller and more uniform in the whole working area of the pad after long time machining [1]. On the other hand, it has also been shown that the microroughness measured by AFM became worse as the structural uniformity of the pad became worse, which was evaluated from the uniformity of light transmission through the pads [2]. These facts indicate that a commercial polishing pads which gives higher total flatness gives worse microroughness. Generally to say, new pads suitable for the primary polishing should be developed which have both high instantaneous elasticity and high structural uniformity.

2 Advanced polishing process

As clearly recognized from surface smoothing models illustrated in Fig. 1, the initial rough surface will generally become smoother and flatter with smaller stock removal as the hardness of the polisher becomes higher. In the conventional polishing of Si wafers, very hard polishers could not be used because no remaining of subsurface damage was allowed. On the other hand, hard polishers can be used in the mechanochemical polishing the authors have developed [3], in which mechanically soft and chemically reactive abrasives are used, so that higher quality wafers can be prepared.

In mechanochemical polishing of Si wafers, BaCO_3 can be effectively used as the abrasive, which is mechanically much softer than silicon and easily reactable with thin silicon-oxide layer existing on the wafer surface [4]. Si surfaces can be polished much sooner and more smoothly with BaCO_3 abrasive than with the silica slurry which is widely used in production line [5].

Harder polisher can be used in BaCO_3 abrasive polishing, as the soft powder is expected not to indent into the wafer surface and to remain no mechanical damages. A porous Teflon disk has

been tested as one of the hard polishers. This new hard polisher has brought extremely high flatness about 1/3 of the NTV value obtained by the conventional soft pad. The surface roughness of $R_{\max}$ 8 nm which is better value than that of 10 nm obtained by the conventional polisher was also realized by the Teflon polisher.

3 Possibility of ultra-precision grinding as a future Si wafer machining process

In order to keep the shape accuracy of the pre-machined (lapped and etched) wafers or to diminish the shape degradation in polishing process, stock removal in polishing should be minimized. After conventional lapping and etching processes, usual wafers have the surface roughness $R_{\max}$ over $0.2 \sim 0.3 \mu\text{m}$. When such a rough surface is polished by a conventional soft polisher, much stock removal should be required for obtaining a smooth surface as supposed from Fig. 1(a). In order to minimize the stock removal in polishing, harder polisher should be used as indicated in Fig. 1(b) and (c) and/or the surface roughness before polishing should have been minimized. As one of the most effective machining processes to realize this requirement, ultra-precision ductile mode grinding should be introduced.

In ultra-precision grinding technology, machining performances would be related with the error or accuracy-degradation factors as described in Fig. 2. If all of the scattering errors are prevented from exceeding more than the Dc value, crack-free ductile mode grinding will be realized, as Miyashita has proposed [6]. The 'Dc value' means the critical depth of cut under which brittle materials can be machined in ductile mode with no brittle fractures. In case of Si single crystal, the Dc value is estimated at about $0.1 \mu\text{m}$ from theoretical calculations on crack generation criteria based on deformation simulation in static indentation of a pyramidal diamond indenter [7].

The authors have developed an ultra-precision grinding machine, which has a wheel head feeding mechanism of the minimum step of 10 nm and the high loop-stiffness over $15\text{kgf}/\mu\text{m}$ [8]. By trial grinding of Si wafers by this machine, high TTV flatness better than $0.6 \mu\text{m}$ for 6 inch wafers and a crack-free surface ground in ductile mode were realized.

As a concluding remark, a new machining process suitable for production of extremely high grade Si wafer will be proposed, in which ultra-precision grinding is adopted as the main process followed by light final polishing. The flow-chart of the future machining process compared with the conventional one is shown in Fig. 3. As a further ultimate process, a machining system would be established, in which the final polishing be sequentially carried out in the same grinding system.

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Table I Processing values required for DRAM wafers

	16MDRAM	64MDRAM	256MDRAM
Wafer size (inch)	8	major 8 minor 12(10)	major 12(10) minor 8
Min. pattern width (μm)	0.5	0.3	0.2
TTV (μm)	<3.0	<2.5	<2.0
LTV (μm)	<0.5 [20mm ^L]	<0.3 [25mm ^L]	<0.2 [30mm ^L]
Chip area (mm ²)	ca. 120	ca. 200	ca. 330
Min. particle size (μm)	0.10	0.06	0.04
Max. number of particles/wafer	5	~1	not clear yet
Sample production	'90	'93	'96
Commercial production	'92	'95	'98

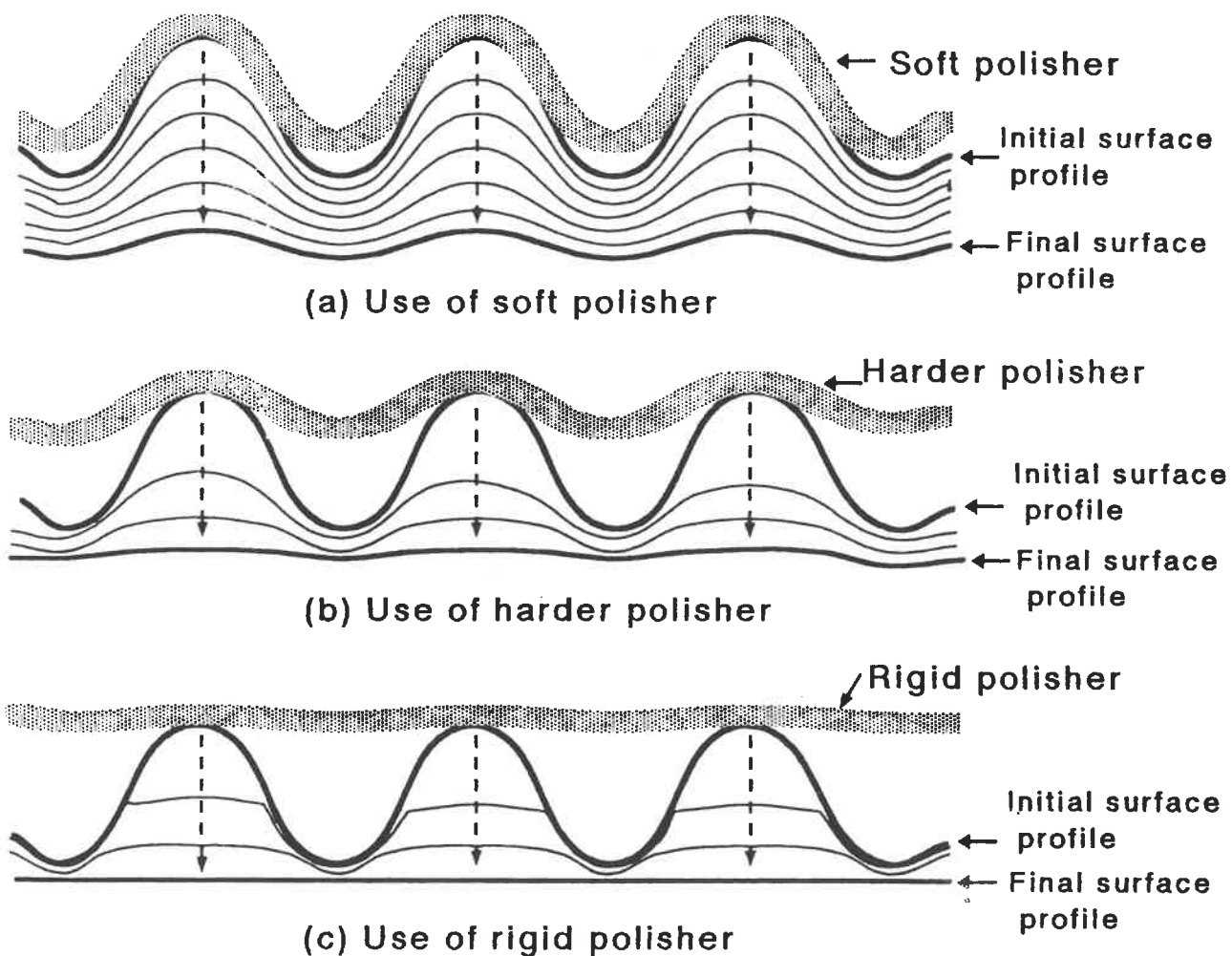


Fig. 1 Surface smoothing models in polishing

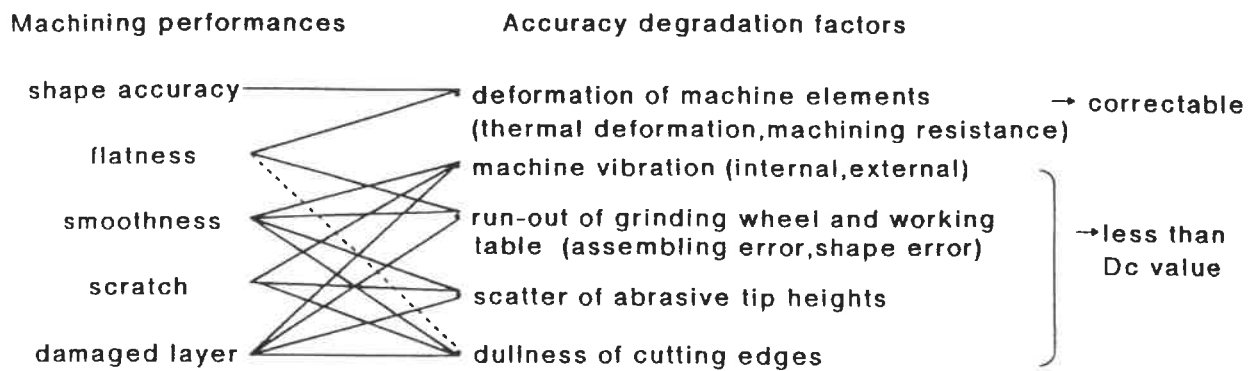


Fig. 2 Relation between machining performances and error and accuracy-degradation factors in ultra-precision grinding

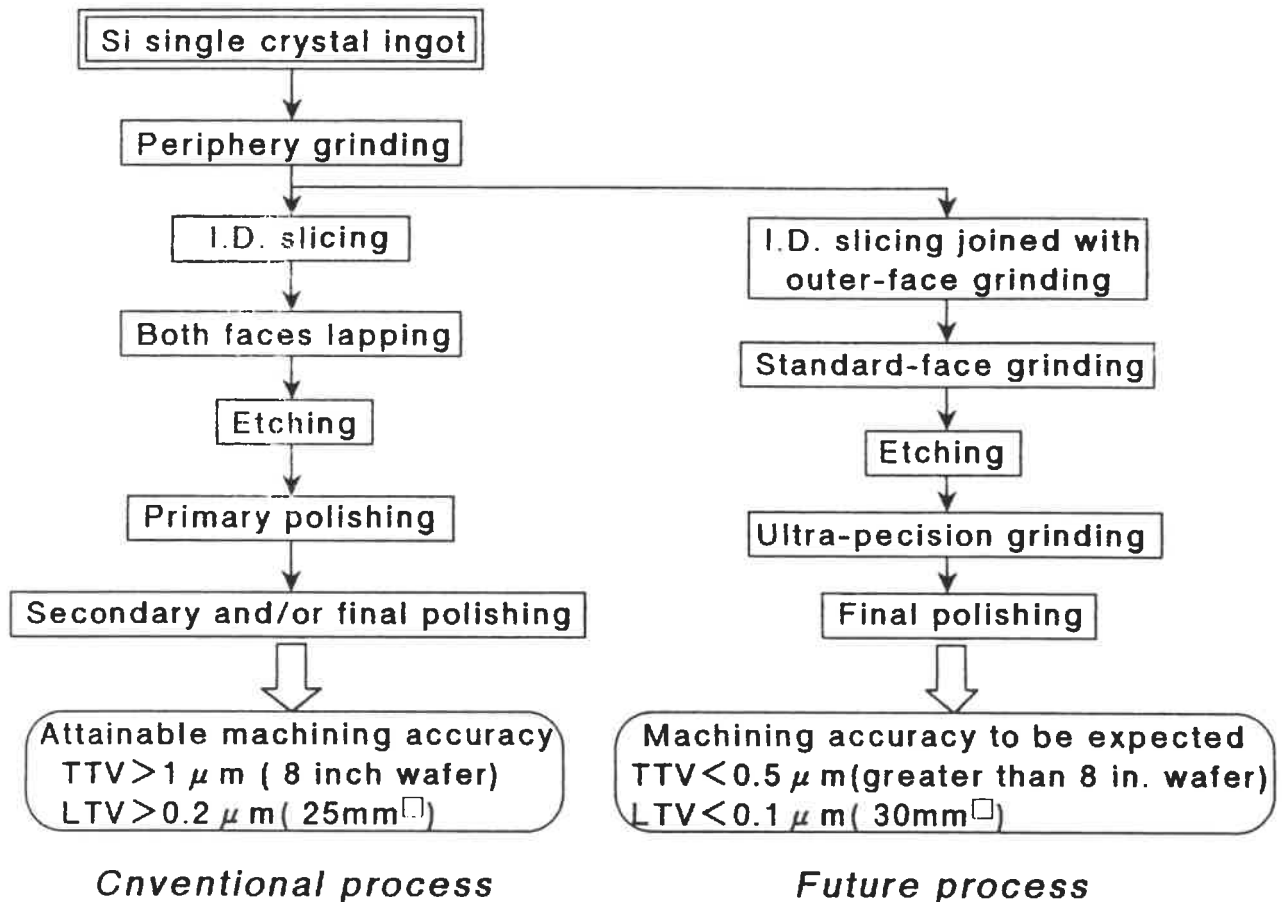


Fig. 3 Si wafer machining process in the present and the future

A full-aperture optical test for large convex aspheric mirrors

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A technique for testing large, highly aspheric convex secondary mirrors is being pursued at the Steward Observatory Mirror Lab. This new test uses full aperture test plate with a computer-generated hologram (CGH) fabricated onto a concave spherical reference surface. Fringes of interference are viewed through the test plate, which is supported several millimeters from the secondary. The hologram consists of annular rings of metal spaced at intervals as small as 80 μm and as large as 500 μm . The accuracy of the surface measurement using this technique is expected to be better than 60 nm P-V and 10 nm rms for testing mirrors with up to 331 μm departure from the best fit sphere. Considerably higher accuracy can be attained for the measurement of less aspheric surfaces.

The holographic test plate is attractive for testing convex aspheres for several reasons listed below:

1. Inherent accuracy. The accuracy of the test is limited only by the quality of a concave spherical surface and the accuracy of the ring locations. Both may be verified to high accuracy.
2. High testing efficiency. The test allows a full-aperture measurement with a small central obscuration. Once the test facility is in place, this test will allow rapid testing to very high accuracy. Vibration and atmospheric effects that often make interferometric testing difficult will be negligible for the holographic test plate.
3. Moderate cost. There is no inexpensive way to test convex aspheres to high accuracy. Testing with a holographic test plate is likely to cost less than conventional optical tests or mechanical profilometry.

The technique of testing aspheric surfaces with a holographic test plate may be used for the measurement of any aspheric surface. The test can be optimized to give complete stray order rejection while matching beam intensities to obtain high contrast fringes.

Convex spherical surfaces are commonly tested with matching concave test plates. When the test plate is supported next to the convex optic and illuminated with sufficiently coherent light, fringes of interference show the shape difference between the two parts. The concave sphere may be tested independently from its center of curvature, so this shape difference is used to measure the convex surface. In a similar manner to the test of convex spheres, aspheric surfaces may be tested using aspheric test plates.

The holographic test measures an aspheric surface using a spherical test plate with a computer-generated hologram fabricated onto the reference surface. The diffraction from the hologram forms the reference wavefront and the reflection from the secondary forms the test wavefront (See Fig. 1). The test plate is illuminated with light that refracts from the reference sphere to strike the secondary mirror at normal incidence. This light reflects back onto itself to form the test beam. Any figure errors in the secondary mirror will be imparted to this test wavefront. The reference wavefront is formed by diffraction from the ring pattern on the reference sphere. The CGH is designed to make this reference beam match the test wavefront so it also retraces the incident path. (This is known as a Littrow configuration for a grating.) To perform a test, laser light from a point source will be used so the desired wavefronts will converge to a conjugate image point, where an aperture will block undesired orders of diffraction.

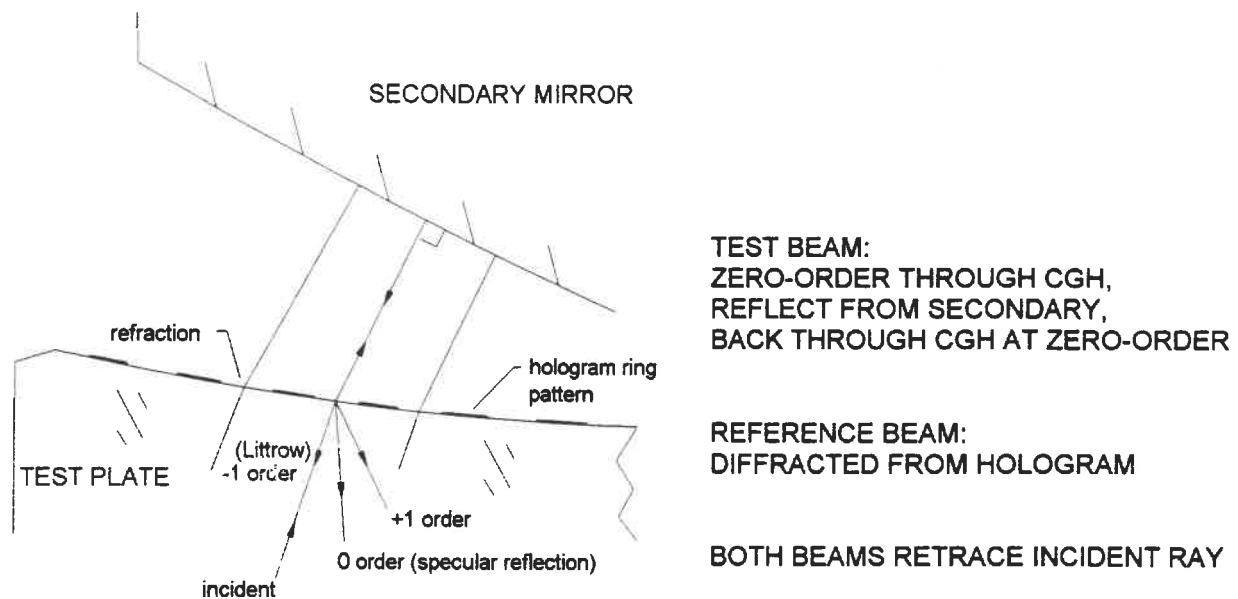


Figure 1. Reference and test beams for a holographic test plate measurement. The test plate has a reference spherical surface with a ring pattern drawn onto it. A reference beam diffracted from the hologram interferes with the test beam reflected from the secondary to allow an interferometric test. By shifting one surface with respect to the other, phase shifting interferometry is used.

Additional optics are required to illuminate the test plate such that the rays will be normally incident to the mirror being tested. Low quality illumination optics can be used without degrading the test accuracy because the reference and test beams are coincident and equally affected by the illumination system. Only the difference between the two wavefronts is measured. This fact allows the test to be economical, because the requirements on the large optics, including the refractive index variations in the test plate and local seeing and vibration, are quite loose. Only the reference spherical surface of the test plate must be figured and measured accurately. That surface is a concave sphere, which is relatively easy to fabricate and measure.

Testing with a CGH may be thought of in terms of a moiré effect. The CGH is a binary representation of the expected fringe pattern formed by an aspheric test beam and a reference beam. When the live interference pattern is superimposed on the CGH, a moiré, or spatial frequency "beating" effect is observed. When properly filtered in the frequency domain, the moiré fringes directly give the shape difference between the two wavefronts, thus the shape error in the asphere.

Rather than using tilt in the CGH to separate the diffracted orders, the circular symmetry of the test is preserved. The ring pattern uses power to separate the orders, causing them to come to focus at different axial positions (See Fig. 2). A small aperture is placed at the position where the desired order comes to a sharp focus, allowing only that order to pass. By optimally choosing the amount of power in the CGH and the size of the stop, the leakage of stray orders is limited to the central *unused* region of the mirror that is ignored during testing. If slope errors in the illumination optics are bad enough, they could deflect the correct order out and a stray order into the aperture. The holograms are designed to allow slope errors in the system of up to ± 2 mrad without causing any order leakage.

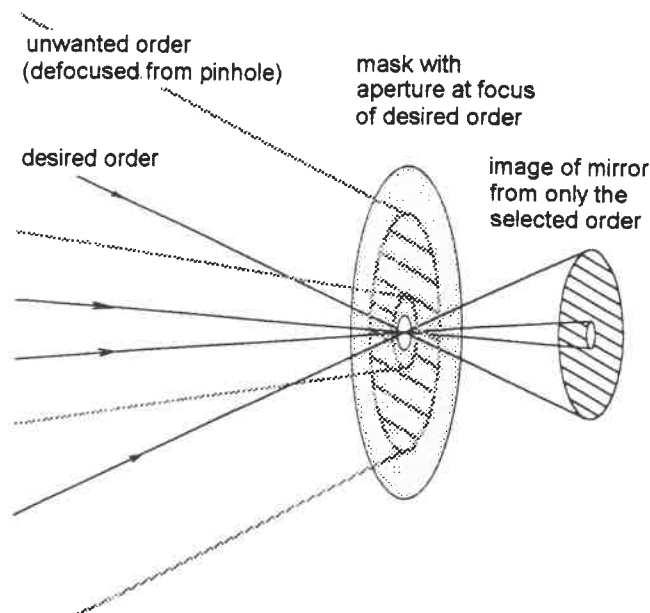


Figure 2. Rejection of stray diffraction orders. The order rejection relies on two principles. (1) The desired order comes to a sharp focus where all other orders are out of focus, and (2) An annular pupil is used. There is a central untested region that is blocked elsewhere.

The accuracy of the wavefront created by CGH is directly related to the position accuracy of the bands. The error in the diffracted wavefront is simply

$$\frac{\Delta W}{\lambda} = \frac{\Delta x}{s}$$

Where Δx is the error in band position, s is the local band spacing. The CGH test plate may have spacings as small as 80 μm so an accuracy of 5 μm in the band position will guarantee $\lambda/16$ wavefront quality.

There are several feasible methods currently being studied to fabricate a rotational CGH onto a curved surface. These methods generally use a precise rotary stage to spin the test plate under a focused optical beam (See Fig. 3). Linear stages control the radial and axial position of the writing head. The stages need only several-micron accuracy, which is readily available in coordinate measuring machines. The pattern may be drawn by exposing photoresist, ablating a metallic coating, or by creating a thin oxidation layer by heating with a focused laser.

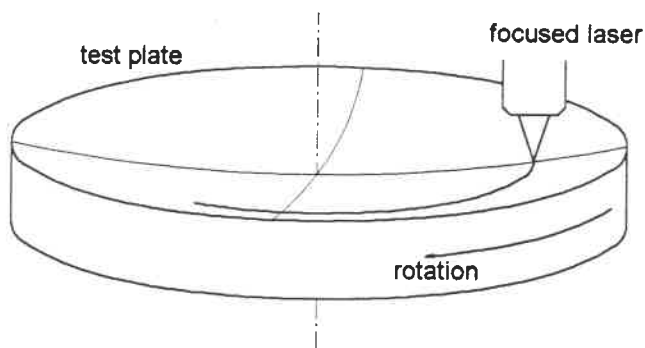


Figure 3. General geometry for optically writing ring patterns onto curved test plates. A focused laser beam is moved radially to expose one ring at a time onto the rotating optic. The line width is varied by going out of focus.

An experiment was performed to verify the technique of using a test plate with a CGH drawn on the reference surface. The test was simplified by using flat surfaces and linear rulings, but the ruling pitch, illumination and imaging geometry, and measurement using phase shifting interferometry simulated the optical test of secondary mirrors. This test, shown in Fig. 4, verified that optical testing with a CGH test plate gives high contrast fringes that are insensitive to vibration and air currents. A 200- μm pitch ruling with distortion errors expected to be several microns was used to measure the surface figure of a small flat to less than 2 nm rms (See Fig. 5.).

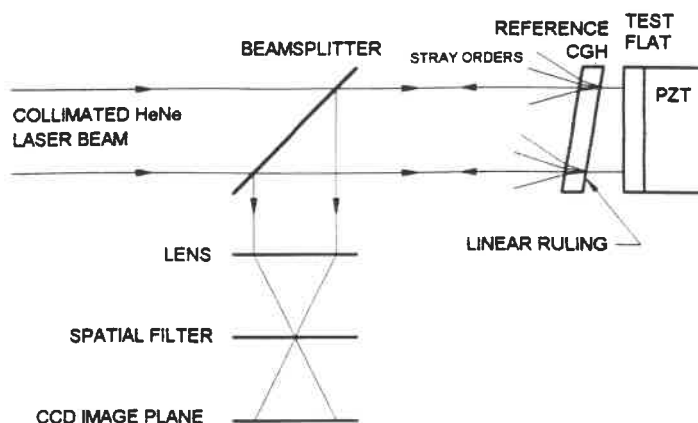


Figure 4. Optical layout of the setup used to verify the use of a CGH test plate.

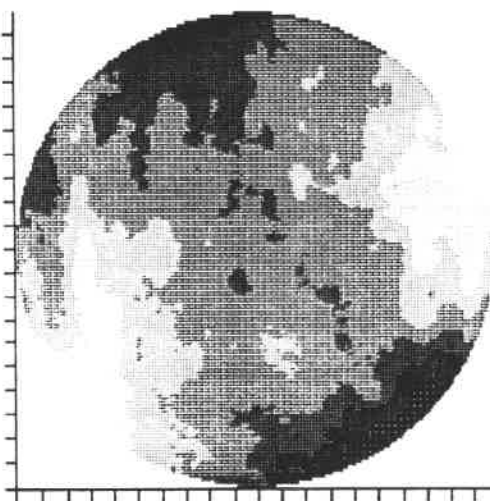


Figure 5. Error in measuring test flat with CGH test plate. The figure errors are 2.09 nm rms and 13.6 nm P-V. The CGH distortion was determined to cause 1.92 nm rms figure error, and the noise in the measurement was determined to cause 0.54 nm rms error. The contours shown are at 2 nm intervals.

The duty cycle of the hologram, defined as the ratio of the line width to the center-to-center spacing is chosen to match the intensities of the reference and test beams, giving high contrast fringes. A duty cycle of 22% would be used for chrome rings on bare glass.

A CGH test plate technique is a highly accurate and efficient method for testing large convex aspheres and other aspheric surfaces. The fabrication of the holograms, which consist of annular rings of metal, can be performed to machine-tool accuracy resulting in a test with nm accuracy. By optimizing the test geometry, the hologram metal, and the ruling duty cycle, the test can give surface data with a high signal-to-noise ratio over the entire tested region.

Near-infrared phase-modulation interferometer for surface flatness measurement

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High-quality metrology is a critical aspect of precision machining, grinding and polishing of flat glass and metal parts. The close tolerances encountered are often in the range of the resolution of optical interferometry, but surface roughness and departure can frustrate the imaging properties of common visible-wavelength instruments.

We have found that a modest increase in source wavelength to the near infrared greatly extends the range of usefulness of optical interferometry in many precision engineering applications. Implementation of this technology has been facilitated by improvements in laser diode design and packaging for the communications industry, which requires 1.3 μm and 1.55 μm radiation for transmission of information via optical fiber. These wavelengths are capable of transforming surfaces that appear very rough to a 0.63 μm HeNe laser into nearly specular surfaces. A further benefit is the reduced fringe density for large slopes, which increases the operational range of the instrument for a given camera resolution. Finally, an important characteristic of 1.55 μm radiation is that it is eye-safe.

Our experimental laser-diode interferometer uses the off-axis Fizeau geometry shown schematically in Figure 1. An off-axis arrangement was chosen to avoid problems with optical feedback to the laser diode. Other advantages to this geometry include the elimination of the expensive beam splitter prism and quarter-wave plate combination characteristic of most Fizeau interferometers, and the reduction in the number of parts contributing to spurious reflections and scattering. To further simplify the optical design, spatial filtering and beam-shaping optics for the laser have been left out. Instead, we use the natural divergence of the beam emitted from a single-mode fiber for illumination. The long f-ratio imposed by the rate of beam divergence permits the use of a very simple collimator optic consisting of a single f12 plano-convex lens oriented so that the plano side doubles as a the reference surface. The camera is an IR vidicon with 600-line resolution.

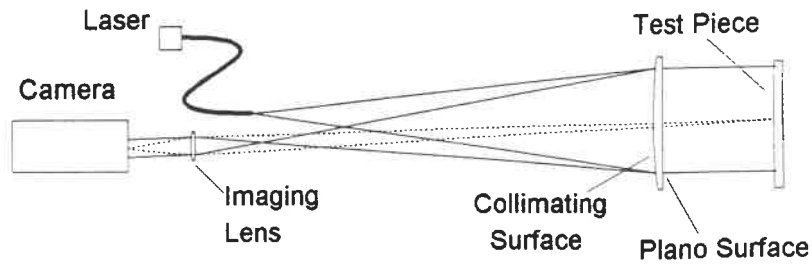


Figure 1. Near-infrared laser diode interferometer for flat surface testing.

A useful characteristic of laser diodes is that the emission wavelength can be modified very easily by adjusting the pump current. Wavelength tuning is an effective way of performing phase shifting without the need for large and expensive piezo stages.^{1,2,3} We acquire a succession of five frames at phases separated by $\pi/2$ and calculate the phase at each pixel in the image using an algorithm described by Schwider⁴ and by Hariharan.⁵ The resulting phase maps can be post-processed to generate three-dimensional images.

The phase map shown in Figure 2 illustrates the large slope acceptance of the interferometer. The sample is the front surface of an automobile rear-view mirror. This is a highly distorted optic from the point of view of interferometry and cannot be measured with conventional HeNe-based instrumentation. The $1.55\mu\text{m}$ wavelength more than doubles the size of the interference fringes, so that physical slopes of more than $1\mu\text{m}/\text{mm}$ in this sample are easily accommodated.

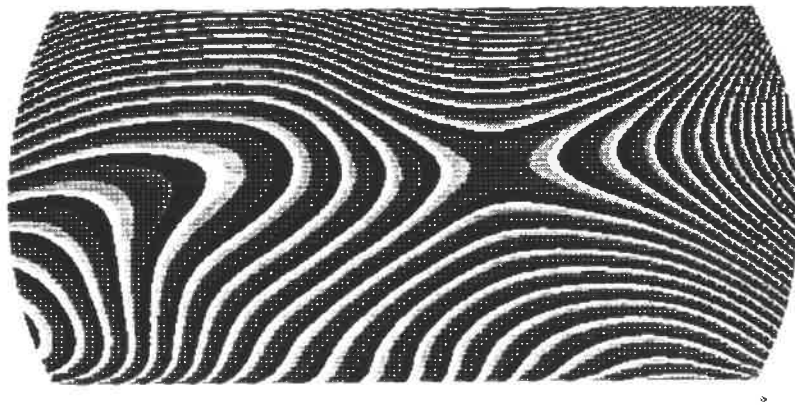


Figure 2. Wrapped phases from an automobile rear-view mirror.

Figure 3 shows the phase map of a 90mm-diameter aluminum hard-disk blank taken with the $1.55\mu\text{m}$ interferometer. The blank has a wire-brush finish and an RMS surface roughness of $0.3\mu\text{m}$. This sample is far too rough for interferometry at HeNe wavelengths and is a good example of what can be achieved in the near infrared. The plot shown in Figure 4 was obtained by connecting the phases in Figure 3, fitting the first 30 Zernike coefficients to the data and finally generating a representation of the surface mathematically.



Figure 3. Wrapped phases from wire-brush finished aluminum disk.

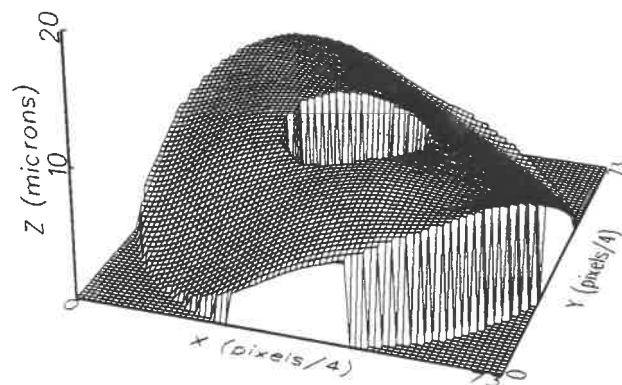


Figure 4. Profile obtained from phase map in Figure 3.

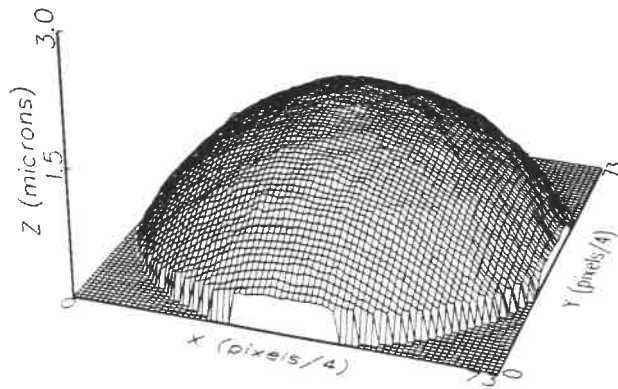


Figure 5. Height map for a 75-mm diameter ground glass surface.

Another use of the near-infrared interferometer is the characterization of optics during fine-grinding. Figure 5 shows the residual figure of a 75-mm diameter glass flat that has been ground with 5 μ m grit. The fringes for this piece are well defined at 1.55 μ m and are easily processed using conventional software. It goes without saying that it is hopeless to try and measure untreated ground-glass surfaces with a conventional visible-wavelength interferometer.

These examples illustrate how a modest increase in wavelength to the near-infrared can significantly enlarge the range of applications for phase-shifting interferometry. For precision machining and optical applications requiring high accuracy with improved tolerance for surface roughness and departure, the 1.55 μ m laser-diode interferometer may be ideal.

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Tunneling Current Characterization: An Application of an X-ray Interferometric Calibrated Flexure Stage.

J. A. Miller^a and S. T. Smith^b

Introduction:

Accurate measurements of surface microtopology with a scanning tunneling microscope (STM) requires a knowledge of the tip-sample interaction. In order to ascertain the accuracy of measurements, the repeatability of current versus displacement, $I(S)$, characteristics must be determined. Also, there are devices that have been conceived which operate with a tunneling sensor¹ as the position sensitive transducer. Among these devices are patents for a magnetometer², a tunneling microphone², and a nanomemory³ design. For design purposes, each of these devices require an accurate knowledge of the operation of the tunneling transducer. A device is described for obtaining accurate calibration $I(S)$ characteristics and results of the measurement of the current between a planar sample and sharp tip⁴ are given.

Design:

A schematic illustration of the device used for obtaining $I(S)$ characteristics between a tunneling tip and a conducting surface is shown in Figure 1. A flexure stage, machined from a monolithic piece of silicon, was electromagnetically actuated to provide the tip-sample displacement. An x-ray interferometer, also machined into the single crystal silicon, provided displacement calibration of the flexure stage over a range of 20 nm, with an accuracy of better than 50 pm. The EM actuator has been shown to provide a linear force that is directly proportional to current and to a first order approximation is independent of magnet position⁵.

The tunneling tip, made from mechanically sharpened 0.25 mm diam

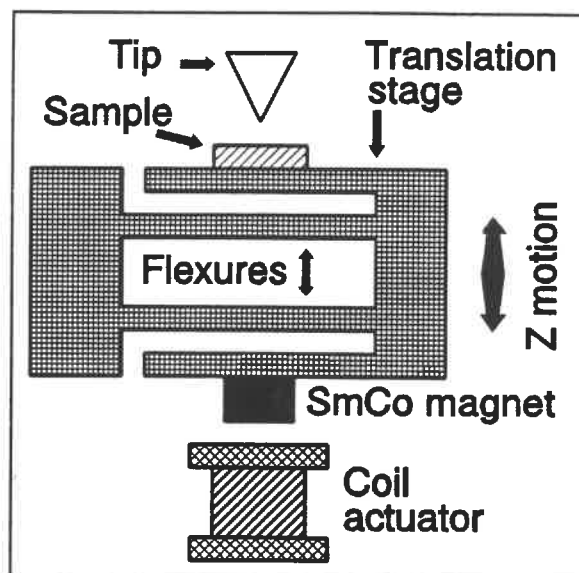


Figure 1. Schematic showing the flexure operation of the $I(S)$ stage.

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wire, was held by friction in a vertical tube mounted in a coarse positioning screw as shown in Figure 2. The coarse screw is used to adjust the tunneling tip to within a few micrometers of the sample surface. The screw is mounted in a steel nut attached to the bottom of the Zerodur. A free nut on the opposite side, in conjunction with two compliant nylon washers, provides a preload force which stabilizes the tip-holder and prevents the two nuts from locking the screw.

The screw also provides an electrical connection through a $1\text{ M}\Omega$ resistor to ground. A -40 mV bias was placed on the sample which induced a current when the tip and sample were sufficiently close. Since the tunneling currents are on the order of 2 nA , the voltage drop across the resistor was about 2 mV . This voltage drop was amplified 1000 times by an op-amp circuit to obtain a current to voltage output of 1 V nA^{-1} . The noise level was no more than 20 mV which corresponds to a current of 20 pA .

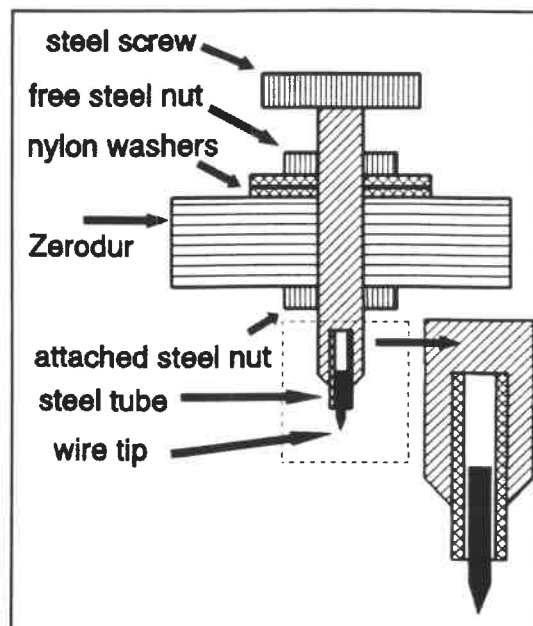


Figure 2. Tunneling tip and coarse positioning screw mount.

Fine-approach of the tunneling probe was achieved using the Zerodur reduction lever mechanism shown in Figure 3. The complete assembly rests

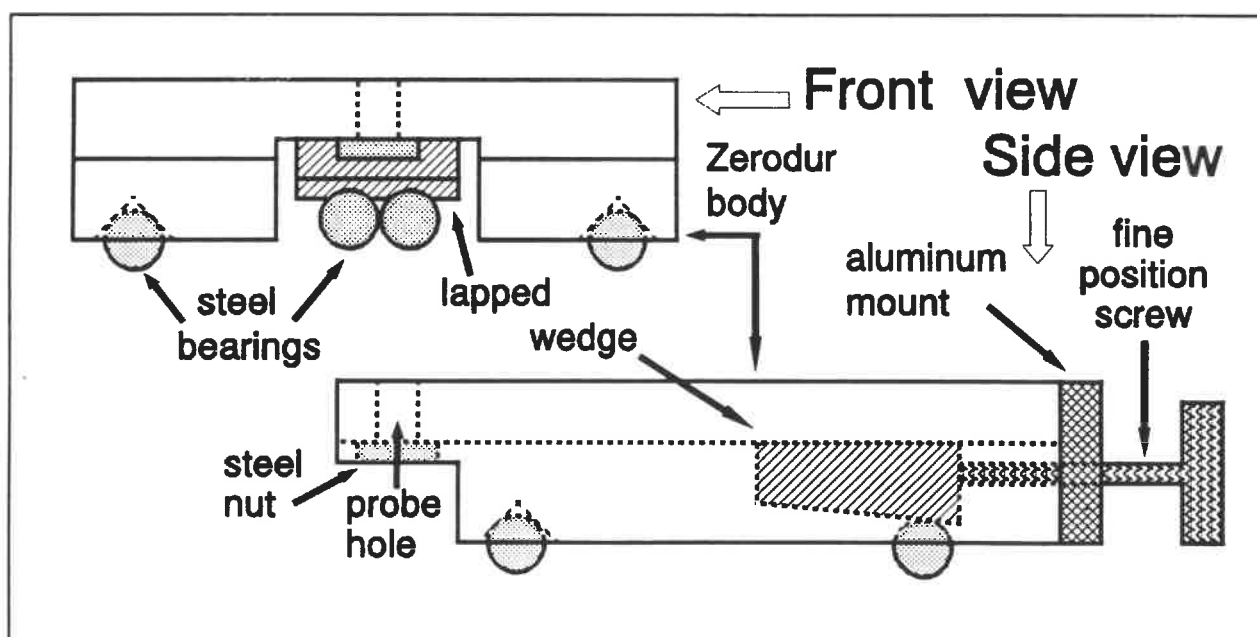


Figure 3. Diagram of the wedge-lever mechanism.

directly on the top of the interferometer flexure stage through a steel ball bearing kinematic mount. To the rear of the body is a fine position screw of 0.5 mm pitch. This is used to drive a lapped wedge with a 100:1 aspect ratio, which is further attenuated by a 5:1 reduction about the pivot. The entire mechanism provides a tip displacement of 1000 nm for each screw revolution. Using this system open-loop, manual adjustment of 15 nm steps was readily achieved.

Performance:

The low expansion ceramic Zerodur body along with the use of similar materials for tip holder and the kinematic mount (steel) provided excellent thermal stability. Figure 4 contains six I(S) plots of the same tip-sample combination and illustrates two points. First, the plots illustrate that the I(S) characteristics are not consistent, but vary in shape. Second, the thermal stability of the measurement device is verified with the curves shifting less than 1 nm over a 15 min interval. Additionally the tip has been observed to stay within the 20 nm range of the flexure for over 4 hr.

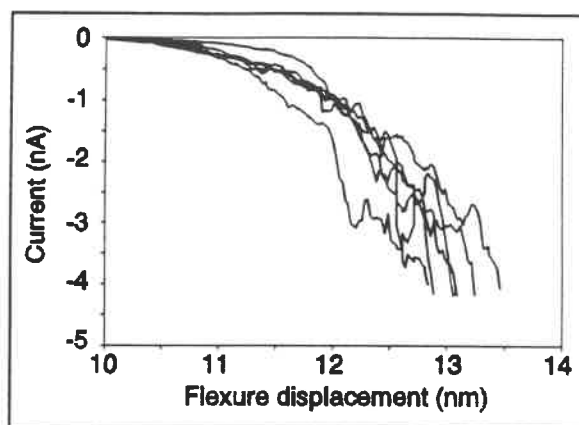


Figure 4. Six I(S) curve taken over a 15 minute interval.

Results:

Data was collected in air ambient using a highly oriented pyrolytic graphite (HOPG) sample and a variety of tip materials. For small biases between the tip and sample, simple tunneling theory⁶ indicates that the current should depend exponentially on the separation with the slope of the natural log of the current being constant. The exponential nature of

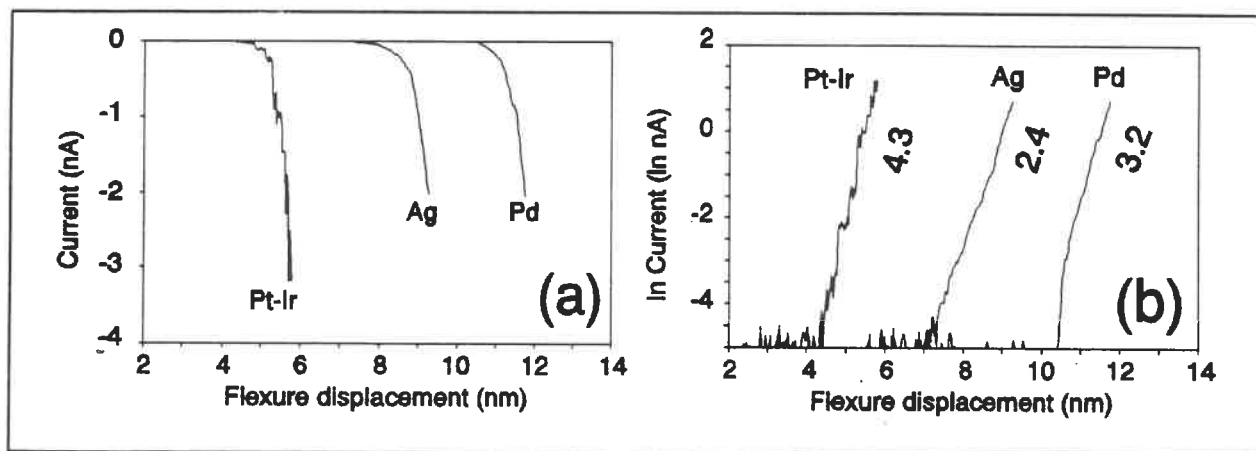


Figure 5. (a) shows I(S) data taken with an HOPG sample and Pd, Ag, and Pt-Ir probes. (b) shows the natural log of the data in (a). Slopes are shown for each of the natural log curves in nm^{-1} .

tunneling current against relative displacement between the sample and tip can be seen in Figure 5(a), however Figure 5(b) reveals that the slopes of the natural log plots were found to be considerably lower than the theoretical value of 23 nm^{-1} for HOPG and Pt. Since Pt and graphite are two materials that are inert in air, that is, the air does not substantially react with the materials' surfaces thus changing the electron barrier, the combination is expected to produce the most consistent $I(s)$ data. Also, the experimentation showed that the $I(s)$ curves were similar but not repeatable for each individual tip, and distinct for different tips even of the same material.

Conclusion:

Since the curves for a single tip are found to vary over the 15 min duration, and the $I(S)$ characteristics vary for each individual probe, positioning the tip at a given distance from the surface in air using the current as a repeatable servo reference may only be possible by knowing the tip geometry to within a significant scale, and with surfaces having well defined electronic properties. This may not be possible in air.

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²Wandass, Joseph H., Colton, Richard J., and Murday, James S., "Magnetic field sensor and device for determining the magnetostriction of a material based on a tunneling tip detector and methods of using same", U. S. Patent #US5103174.

³Quate, Calvin F., "Method and means for data storage using tunnel current data readout", U. S. Patent # US4575822.

⁴For a report on a vacuum tunneling device using spherical gold electrodes see Teague, E. Clayton, "Room temperature gold-vacuum-gold tunneling experiments", *J. Res. NBS*, **91**(4), 170 (1986).

⁵Smith, S. T., and Chetwynd, D. G., "An optimised magnet-coil force actuator and its application to linear spring mechanisms", *Proc. Inst. Mech. Eng.* **204**(C4), 243 (1990). Chetwynd, D. G., Cockerton, S. C., Smith, S. T., and Fung, W. W., "The design and operation of monolithic x-ray interferometers for superprecision metrology", *Nanotechnology* **2**, 1 (1991).

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3.39 μ m INFRARED MULTIWAVELENGTH He-Ne LASER INTERFEROMETER FOR ABSOLUTE DISTANCE MEASUREMENT

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ABSTRACT

This paper describes a new type multiwavelength He-Ne laser interferometer for diameter measurement of large scale work-piece. The laser can oscillate on two wavelength 3.3912 μ m and 3.3922 μ m simultaneously. By switching between two equal intensity state it provides a two order wavelength hierarchy pyramid from 1m to 3.3912 μ m. The range of this system is up to 20m and the accuracy $5\mu\text{m}+1.5\cdot 10^{-6}L$. (L is the measured distance)

1. INTRODUCTION

Using an optical wavelength to measure absolute distance is the classical experiments of measuring the international standard meter. At present the multiwavelength lasers have been used to measure long distance because of their superior coherence and precision frequency stability. Among them the CO₂ laser is in common use. It can oscillate on a large number of closed interval lines, and form a wavelength hierarchy from 25m to 10.6 μ m, after an efficient application of the excess fraction reduction method by Tilford.

Recently He-Ne laser becomes a useful source for absolute distance measurement gradually. The National Institute of Metrology in China has developed a beat-wave interferometer with a new type Zeeman He-Ne laser which has two resonating modes with 1080MHZ frequency difference. The paper describes a new laser interferometer employing the dual wavelength He-Ne laser which can form two degree wavelength hierarchy pyramid. The interferometer measures the phase of the beat signal coming from first order differential wavelength, and The range of this system is up to 20m and the accuracy $5\mu\text{m}+1.5\cdot 10^{-6}L$. (L is the measured distance).

2. PRINCIPLE OF MEASUREMENT

A entire analysis method to calculate the measured distance by reducing the fraction fringes of every single wavelength in the multiwavelength interferometer was developed by Tilford in 1977. The key point is combining several single wavelengthes to form a wavelength hierarchy pyramid, the higher the order the longer the wavelength, then using it to calculate the true value of the measured distance step by step. Different lasers provide different data reduction process. The method about reducing the limit

to the accuracy of preliminary estimate value about the measured distance and raising the measuring accuracy and extend the measuring range are pointed out also .

Six laser spectrum lines have been observed in He-Ne laser with the Ne atom transimtion from 3S to 3P. Common He-Ne laser only emits the optical wave at 3.3922 μ m wavelength whose gain value is the largest among these six lines. Increasing one line's loss in the laser to check it partly can encourage the other lines to oscillate. Methane(ch4) has selective absorptivity at 3.39 μ m wavelength band, only absorbes 3.3922 μ m wavelength line. When a ch4 cell with the certain length and gas pressure is put into the common laser along the optical axis the laser can emit two wavelength 3.3912 μ m and 3.3922 μ m simultaneously . By selecting the resonant length Lc which meet the equations of (1) and (2), the laser has two equal intensity operation state.

$$L_c = (K+0.5)C/2\Delta V \quad (1)$$

$$L_c < C/2\Delta V_d \quad (2)$$

where ΔV is the frequency difference between 3.3912 μ m and 3.3922 μ m, ΔV_d is the gain curve width. Fig.1 shows the two state wavelength hierarchy pyramid, the two first differential wavelengths λ_a , λ_b and the second synthetic wavelength λ_s .

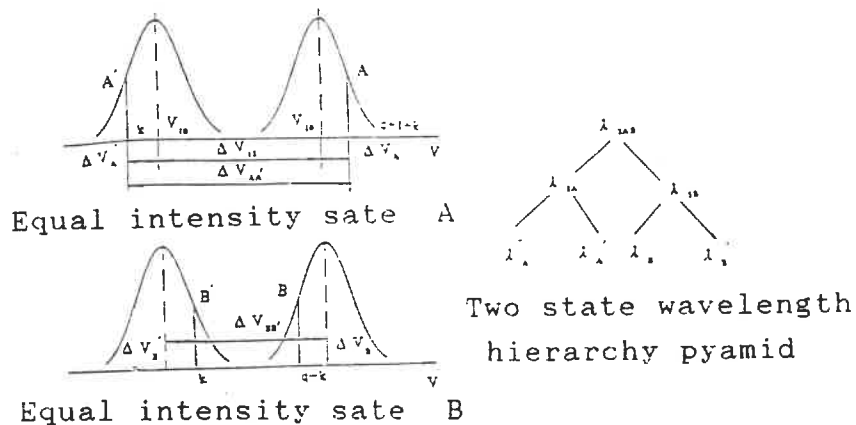


Fig.1. Two state wavelength hierarchy pyramid
According to the In the interferometer theory

$$L = (Ma + \epsilon_a) \lambda_a / 2 \quad (3)$$

$$L = (Mb + \epsilon_b) \lambda_b / 2 \quad (4)$$

where the integral orders Ma and Mb, λ_a and λ_b generate two fraction fringes ϵ_a and ϵ_b . They are unknown. Linear combination of equations (3) and (4) is

$$L = (Ms + \epsilon_s) \lambda_s / 2 \quad (5)$$

where

$$Ms = Mb - Ma \quad (6)$$

$$\epsilon_s = \epsilon_b - \epsilon_a \quad (7)$$

Equation (6) can not define Ms. Once L is coarse measured by some common method such as tape, Ms will be calculated. According to (5) precise L value is get. Using this value to calculate Ma and

M_b , the more precise value of L can be got. But in order to ensure to M_s , M_a and M_b the only values the accuracies of the coarse measurement value and interferometer measurement values must be smaller than the certain threshold values shown in equation (8)

$$|(L_e - L)| < \lambda_s / 4 \quad (8)$$

where L_e is the measurement accuracy values of the preliminary estimate. Because the integral order of every synthetic wavelength is determined on the base of the L values measured by the higher synthetic wavelength the equation (8) is the transition rule of the wavelength hierarchy pyramid. According to Fig 1.

$$\begin{aligned} \lambda_s &= C / (\Delta V_a - \Delta V_b) \\ &= c / \Delta V_c \end{aligned} \quad (9)$$

where

$$\Delta V_c = C / 2L_c \quad (10)$$

ΔV_c is the longitudinal mode interval. L_c is 500mm, λ_s equals about 1000mm. This result is of practical significance since it means it is very easy to use in workshop. λ_a and λ_b are about 11.5mm, much smaller than λ_s . So the measurement accuracy can be higher with an unexcessive precise method to measure the fraction fringes.

3. THE MULTIWAVELENGTH BEAT WAVE INTERFEROMETER

The beat wave interferometer used for measuring the fraction fringes is shown in Fig.2.

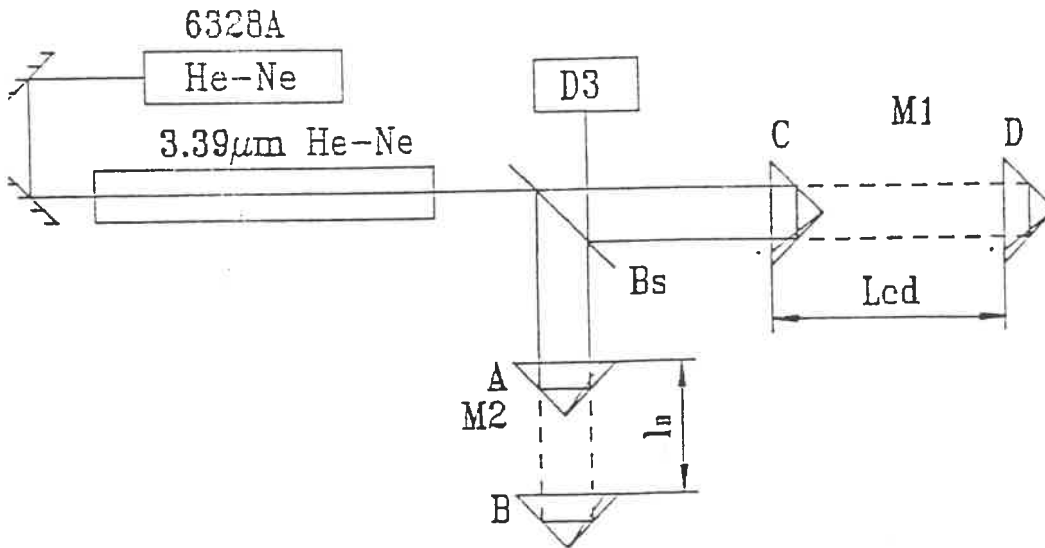


Fig.2 Beat wavelength interferometer.

The optical intensity (I) are detected by the square law detector D1 :

$$I = I_{11} + I_{12} + I_{21} + I_{22} + 2I_{11}I_{12}\cos(2\pi\Delta l/\lambda_1) + 2I_{21}I_{22}\cos(2\pi\Delta l/\lambda_2) \quad (11)$$

omitting the constant terms and combining the alternative terms it can be expressed to

$$I = 2I_1\cos(\pi\Delta l/\lambda_d)\cos(\pi\Delta l/\lambda) + (I_2 - I_1)\cos(2\pi\Delta l/\lambda_2) \quad (12)$$

where

$$\lambda_d = \lambda_1\lambda_2/(\lambda_1 - \lambda_2) \quad (13)$$

(A) of the measured workpiece. By adjusting laser head and the magnetic positioning cell, the laser beam are aimed the reflector which was located the center of the hollow cube corner retroreflector and make the reflected beam back to the laser outlet.

C. By adjusting the spiral in x-direction, the aiming beam is deviated from the reflector and then keep off the aiming beam. At last finishing the measurement of position A.

D. By adjusting the spiral in x-direction, the aiming beam aim the reflector again.

E. Put the magnetic positioning cell attached on the other side (B) of the measured workpiece. By adjusting the magnetic positioning cell, the laser beam aim the reflector and make the reflected beam back to the laser outlet.

F. Repeat the step (C). Finishing the measurement of position B and calculation of the diameter of measured workpiece.

$$\begin{aligned} D &= \text{SQR}[(L+c)^2 + S^2] & (\text{Hole}) \\ D &= \text{SQR}[(L-c)^2 + S^2] & (\text{Axis}) \end{aligned} \quad (17)$$

Where c is constant which can got by measuring a standar length, such as gauge block.

4. ACCURACY OF MEASUREMENT

The measuring accuracy by the second synthetic wavelength λ_s is

$$\Delta L = \Delta \varepsilon_s \times \lambda_s / 2 + L \times \Delta \lambda_s / \lambda_s \quad (18)$$

the measuring accuracy of the fraction fringes is 7×10^{-4} $\Delta \varepsilon_s$ equals 10^{-3} calculated from the equation (7) by the rule of statistical error. $\Delta \lambda_s / \lambda_s$ is reduced from the equation for calculating the synthetic wavelength

$$\Delta \lambda_s / \lambda_s = \lambda_s / \lambda_a (\Delta \lambda_a / \lambda_a) + \lambda_s / \lambda_b (\Delta \lambda_b / \lambda_b) \quad (19)$$

where $\Delta \lambda_a / \lambda_a$ and $\Delta \lambda_b / \lambda_b$ are two differential wavelength stability, they are 1×10^{-6} when the laser is locked to two operation states by the frequency stabilizer. By the rule of statistical error $\Delta \lambda_s / \lambda_s$ equals 1.2×10^{-4} . So ΔL in equation (18) is smaller than a quarter of λ_a when L is within the range of 20M. It is able to be improved by the differential wavelength

$$\Delta L = \Delta \varepsilon_a \times \lambda_a / 2 + L \times \Delta \lambda_a / \lambda_a \quad (20)$$

Where $\Delta \varepsilon_a$ is 7×10^{-4} . The value of first term in equation (20) is large than a quarter of single wavelength, ΔL in equation (20) is the final measurement accuracy.

Considering the affection of correcting the refraction index in air, The measuring accuracy of L equals $5 \mu\text{m} + 1.5 \times 10^{-6} L(\text{m})$.

5. ACKNOWLEDGMENT

This paper introduce An infrared multiwavelength He-Ne laser absolute distance interferometer for diameter measurement of large scale workpiece. Although the measurement accuracy is lower than that of the CO2 multiwavelength laser interfreometer by one order of magnitude, it is enough for most measurement demands workpieces as well as square by some elementary means. At present this technology is successful to apply to measure the diameter of large scale workpiece.

Design strategies for ultraprecision, high stiffness feed system of slide on plain bearing guideways

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1. Introduction

The paper "Design of centerless grinding machine for ductile mode grinding of brittle materials"[1] was presented at the ASPE Annual Meeting 1990. This prototype machine(M-1) was experimentally analyzed thereafter in terms of structural and also metrology frames of machine configuration.

The present paper deals with a commercialized centerless grinding machine(M-2) for ductile grinding of brittle materials based on the improved design concept derived from the research work on the prototype machine M-1. The focus is on design strategies for ultraprecision and high stiffness feed system of slide on plain bearing guideways. The target specifications are a feed resolution of 10 nm and a stiffness in feed direction of 1 N/nm.

An application of plain bearing guideway is useful for designing ultraprecision machine tools insensitive to vibrational environment, because it yields high stiffness and damping in feed direction. However it has not been practically applied to ultraprecision machines as frictional load affects feed resolution. To overcome this problem a positioning system based on the principle of differential force operation[2] was proposed.

2. Prototype machine, M-1

A positioning system designed on this principle and applied to M-1 machine is shown in Fig. 1. A capacitance type feedback sensor has the specifications of the sensitivity of 400 mV/ μm and operating range of $\pm 25 \mu\text{m}$. Differential force operated through a pressure control servo valve and a hydraulic actuator is transmitted to the wheel slide.

Fig. 2 is cross section of the plain guideways of the wheel slide weighing 100 kg. The slide is restricted horizontally and vertically with a series of coupled hydrostatic pad and plain bearing. Both the feed actuator and feedback sensor D are installed at a point off the center of plain guideways.

Fig. 3 shows the layout of several displacement sensors -A, B, C, and D- for detecting the displacement at the indicated several parts of wheel slide. The front side of the slide is measured at the three points -A, B, and C. The sensor D on the bracket supporting the feed actuator is the feed back sensor of the system. The rear side of the slide is measured at the location of the sensor D -point D- in displacement relative to the bracket fixed on the bed.

Feed resolution and motion accuracy are tested in the two cases with the feedback sensor located at D and also B for a consideration of selecting feedback sensor location.

(Case of feedback sensor located at D - rear side of the slide)

Fig. 4 is the records of the outputs of the sensors -A, B and C and the feedback sensor D for the stepwise feed of 10 nm/step $\times$ 10. The sensor A, B, and C at the front side of the slide indicate the displacement of 55.4 to 73.9 nm, while the feedback sensor does just 100 nm same as command. Most part of the lost motions A, B and C can be estimated to be caused by the elastic deformation of the bracket holding the actuator and the feedback sensor D.

(Case of feedback sensor located at B - front side of the slide)

A location of feedback sensor attached to the front side is preferable to be as close as possible to the grinding wheel. In this view point the sensor B at the front side of the head is selected as the feedback sensor in the following experiment. Fig. 5 is the output of the sensors A, B, C and D for the stepwise commands of 10 nm/step $\times$ 10. The feedback sensor B responds precisely to the command, with no lost motion even at reversing points of feed direction. The difference of the outputs A, B, and C is similar to the test results in Fig. 4. On the other hand, the sensor D shows about 1.4 times of the displacement at B, which

is estimated to be caused by the elastic deformation of the bracket supporting the actuator. Total travel at sensors A, B, and C were 2.6, 2.5, 2.3 μm respectively to the commands of 50 nm x 50.

A series of test results on the prototype machine made clear the following design problem ; (1) the configuration of square plain bearing guideways remains inevitably the Abbe's offset between the drive point and the feed load resulting a yawing error of the slide, (2) a deformation of metrology frame, that is the mechanical loop coupling the slide and feedback sensor at the detecting point of the slide displacement is to be less than a target feed resolution of 10 nm. Unfortunately, the structural frame of the system and metrology frame have a common coupling, the bracket fixed on the bed in the prototype machine. The resulted deformation of the bracket from the frictional load was larger than 10 nm.

3. Commercial machine, M-2

The design features of the improved machine developed are ;

(1) application of the plain bearing guideways with double vee configuration which can minimize Abbe's offset between a drive point and guideways, (2) a feedback sensor attached between a grinding wheel head and regulating one, which controls true depth of cut, and (3) a linear scale with 10 nm resolution as a feedback sensor.

The drive point of the slide is located at the center of double vee guideways as shown in Fig. 6, and moreover, the centers of the both wheels are arranged on extended point of this line. Hydrostatic pads can give preload on plain guideways. Feedback sensor has no offset in the yawing error plane, and some offset in the pitching plane, which is less important for this machine.

A linear encoder is installed below blade and between wheels so as to detect directly the distance between the centers of both wheels as shown in Fig. 7. Only the components related to thermal displacement, scale and the attachments are made of low thermal expansion materials. Thus the distance between wheel centers can be kept constant, independent of thermal deformation of machine bed, etc., which makes deviation of work diameter less sensitive to the thermal disturbance. Output sensitivity of NC is 122 mV/ μm and wheel slide weighs 180 kg.

Fig. 8 shows a record chart how the wheel slide moves when command signals of 50 nm/step x 20 are given each every 5 seconds to forward and backward directions. Slide displacements are monitored at points A and B, front side of the slide. Slide moves at both points in accordance with every command signal without lost motion even at the reversing point of feed direction. Yawing of wheel slide there is less than 50 nm/300 mm, which is much improved compared with that of M-1 machine. Detailed chart shows response of slide to the stepwise command of 10 nm/step. Slide makes displacement of some 10 nm/step, equal to one feedback pulse of the scale. Chart shows near 20 nm travel every seven steps which is caused by electrical dividing accuracy of scale lattice distance of 0.55 μm . Fig. 9 shows static stiffness of wheel slide in feed direction when thrust load is applied to the center of wheel spindle. The stiffness is over 1 N/nm.

Fig. 10 is an experimental result of size stability showing little fluctuation of work diameter against time progress. Workpieces are ground intermittently under room temperature environment of ± 2 degrees in centigrade after starting the machine. Diameters stay within 0.5 μm during operating time of one to six hours. It can be said that a new design to monitor the distance between wheel centers to keep it constant is very effective for the stability of work size.

4. Conclusions

From a series of experiments the achieved performances are summarized as (1) relative linear movement between the grinding wheel head and regulating one is controlled exactly in 10 nm/step and the movement can give minimum depth of cut of 10 nm. (2) the stiffness in the feed direction is over 1 N/nm., (3) yawing error of slide is much improved. The yawing caused by reversing feed direction is less than 50 nm/300 mm and no lost motion., and (4) under environmental temperature deviation of ± 2 degrees in centigrade the relative movement between both wheel surfaces is less than 500 nm over 5 operating hours.

1. Yoshioka, J., Miyashita, M., Kanai, A., Daito, M., Hashimoto, F., The ASPE Annual Meeting 1990 in Rochester
2. Kanai, A., Sano, S., Yoshioka, J., Miyashita, M., Nanotechnology 2, 43(1991)

Key Words : ductile mode grinding, ultraprecision grinding machine, slide feed system, plain bearing guideway

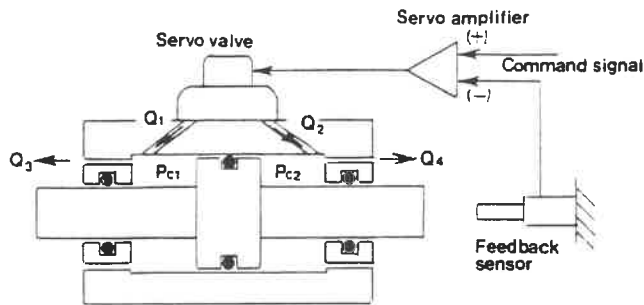


Fig.1 Force operated linear actuator and positioning servo system

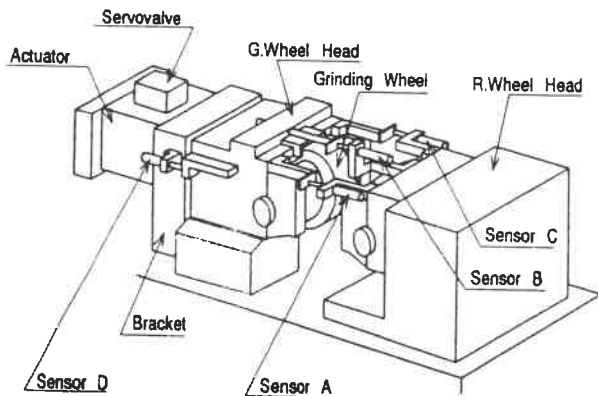


Fig.3 Location of displacement sensors

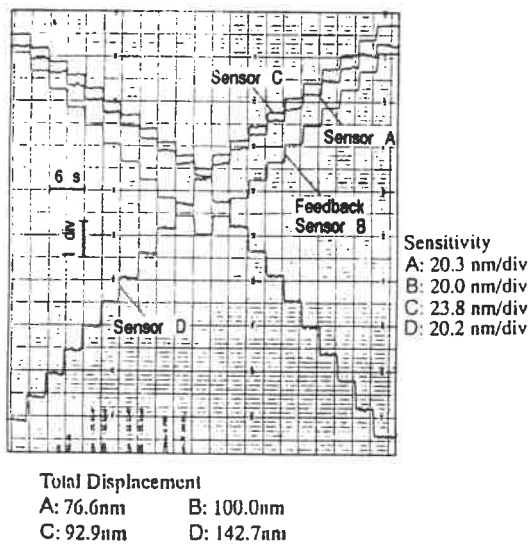
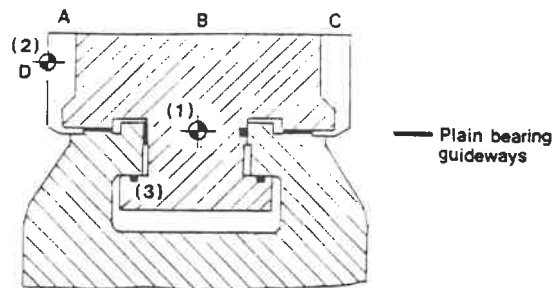


Fig.5 Stepwise feed of 10 nm/step (Feedback sensor : B)



- (1) Center of wheels and feed actuator
- (2) Position sensor
- (3) Hydrostatic pockets (preload)

Fig.2 Guideway mechanism(M-1)

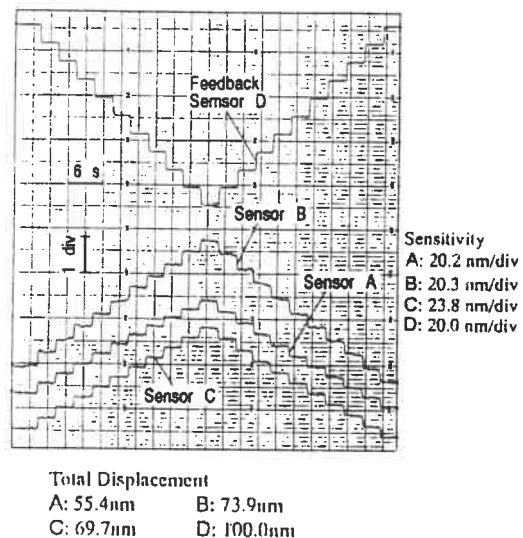
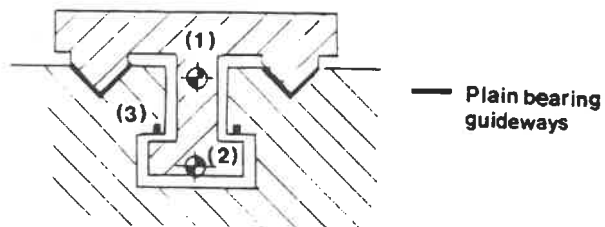


Fig.4 Stepwise feed of 10 nm/step (Feedback sensor : D)



- (1) Center of wheels, guideway and feed actuator
- (2) Linear encoder
- (3) Hydrostatic pockets (preload)

Fig.6 Guideway mechanism (M-2)

Fig.7 Positioning of wheel slide

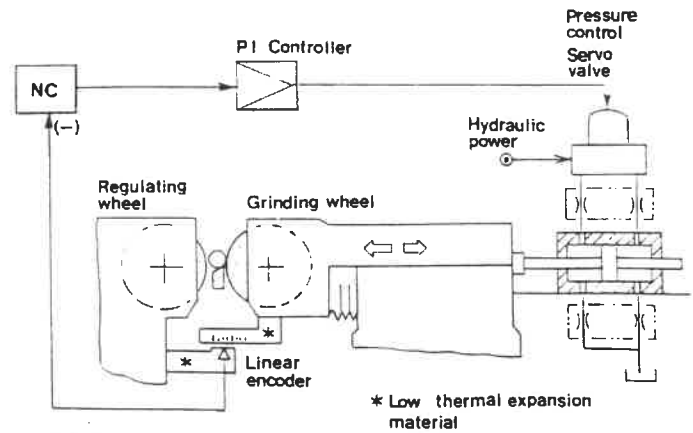


Fig.8 Motion accuracy of wheel slide (50 nm/step)

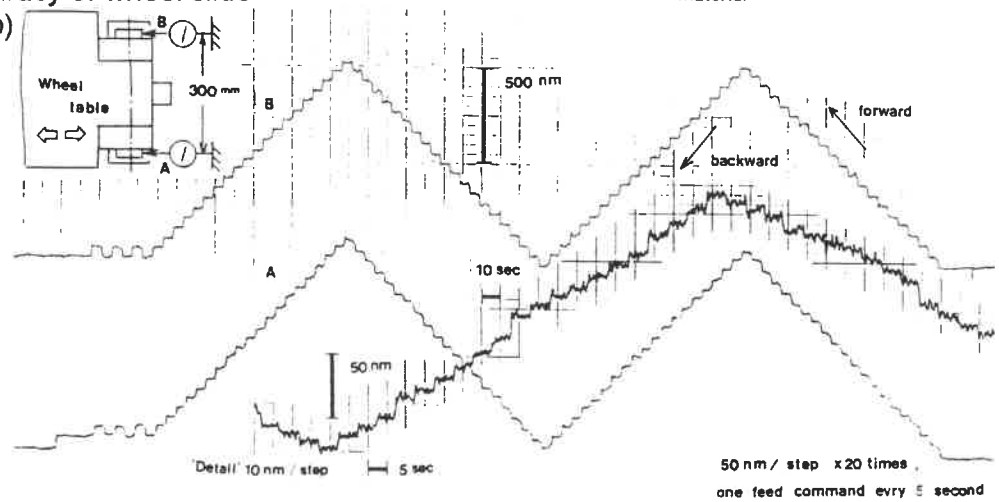


Fig.9 Static stiffness of wheel slide

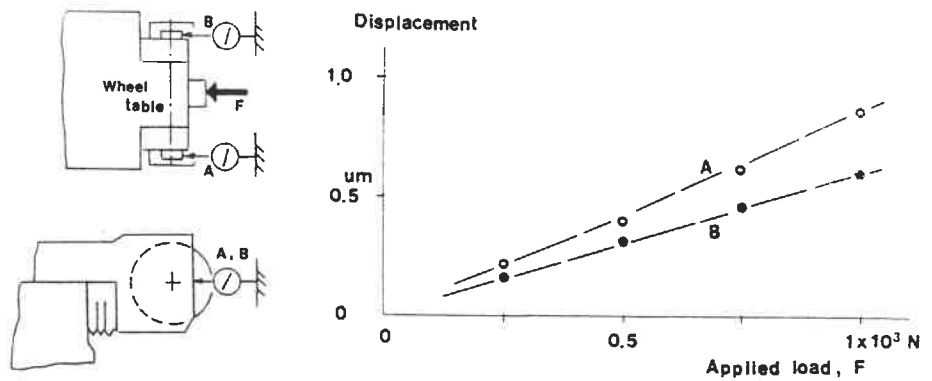
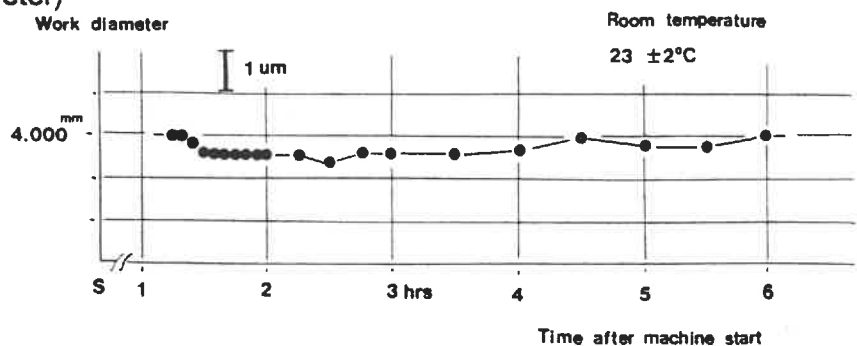


Fig.10 Stability of grinding gap distance(work diameter)



EVALUATION OF A TAPERED ROLLER BEARING SPINDLE FOR HIGH-PRECISION HARD TURNING APPLICATIONS

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INTRODUCTION

Spindles are among the most critical machine components that affect the surface characteristics and the geometry of the parts produced by single point turning. Traditionally, air bearing spindles are used for precision machining such as single point diamond turning. With proper care in the design and installation, air bearing spindles can produce the most accurate axis of rotation with errors in the order of 0.05 micrometers or less. Some limitations with the air bearing spindles are the load carrying capacity and the dynamic stiffness. These limitations are not critical in most single-point diamond turning operations because of the low cutting forces involved. But, in other applications such as precision hard turning, the cutting force and dynamic stiffness requirements may approach or even surpass this limitation. Therefore, other spindle alternatives have to be evaluated for these applications.

In this study, we are evaluate a spindle which consisted of a pair of high precision tapered roller bearings including a HYDRA-RIB^{TM,1} tapered roller bearing. Roller bearing spindles are used for applications that require high bearing stiffness, and can be good candidates for applications such as precision hard turning. For evaluation purposes, we have classified our measurements into four categories: axis of rotation errors, thermal growth, static stiffness, and dynamic stiffness and damping. In this presentation, we will focus on the axis of rotation error measurements.

In addition to evaluating the performance of the spindle, we will show that the axis of rotation error measurements can also be used to help diagnose, analyze and improve the spindle design. The error motions can be analyzed to provide information about bearing defect frequencies, relative contributions of front and back bearings to the error motion, and modulation of bearing defect frequencies. We have identified several relationships between error motions and the bearing design as well as relationships between the critical geometric tolerances of the spindle design and the spindle axis of rotation errors.

EXPERIMENTAL SETUP AND PROCEDURE

Our experimental setup to measure axis of rotation errors in the fixed sensitive direction consisted of 100 mm diameter diamond-turned aluminum test artifact and capacitance probes arranged as shown in Figure 1. Measurements were taken from two locations along the length and two locations on the face of the artifact to evaluate radial, tilt, and axial error motions of the axis of rotation. The analog signals from the probes were fed to a digital scope. To synchronize the data with the angular position of the spindle, the scope

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¹ Commercial equipment or materials are identified in this paper in order to adequately specify certain experimental procedures. In no case does such identification imply recommendation or endorsement by the National Institute of Standards and Technology, nor does it imply that the material or equipment identified is necessarily the best available for the purpose.

recorded data based on the trigger signal from each encoder pulse resulting in 256 points per revolution. Each set of data acquisition was initiated based on top dead center signal (index pulse) of the encoder. Therefore, regardless of the spindle speed and the time the data were taken, each record corresponded to the exact angular position of the spindle and the artifact.

Following the guidelines in the existing standards [1], we started to analyze axis of rotation errors using a sample of data corresponding to 20 revolutions of the spindle. As described in the Axis of Rotation Standard [2], the analysis consisted of calculating and removing of the centering error of the artifact using least squares circle fitting techniques, eliminating artifact profile errors using the reversal technique, and then calculating average and asynchronous components of the axis of rotation errors. However, we observed a significant amount of apparent nonrepeatability in our measurement results. To further study this behavior, we recorded one sample per revolution data over 10,000 revolutions at 650 rpm using a 5.4 Hz antialiasing filter. A cyclic behavior of the asynchronous error due to frequency modulation was observed as shown in Figure 2. This low modulation frequency was caused by the very similar roller cage frequencies of nose and back bearings. In order to include at least one full cycle, we changed the procedure to gather data corresponding to 500 revolutions of the spindle instead of 20 revolutions. We repeated the measurements for three spindle speeds: 100 rpm, 500 rpm, and 650 rpm to evaluate the dynamic effects on the axis of rotation errors. To eliminate the effects of possible instability at each start-up we ran the spindle as much as 20 minutes before we acquired data. We also carried out our tests at two different HYDRA-RIB pressures to identify the effect of bearing preload on the performance.

In order to identify the effects of the geometric alignment between the spindle housing and the shaft, the components were measured on a coordinate measuring machine and a roundness measuring machine before assembling the spindle. The most notable deviation was the parallelism of the front and the rear bearing housing seats, which was approximately 8 micrometers off over 170 mm diameter (about 45 microradians). In spite of this deviation, the spindle was assembled and the axis of rotation error measurements were taken before the corrective machining was done. Then, remachining was done to establish the parallelism within 0.25 micrometers over the same diameter (about 2 microradians). The axis of rotation errors before and after the remachining of the housing were compared.

RESULTS

In order to correlate the axis of rotation errors with bearing design, we carried out Fast Fourier Transform (FFT) analysis on the displacement data of 10,000 revolutions as shown in Figure 3. As we zoom in around 5 Hz of the FFT chart (Figure 4), it clearly shows the peaks that correspond to calculated cage defect frequencies of nose and rear bearings which are 4.9239 Hz and 4.8934 Hz, respectively. The error amplitude caused by the nose bearing is seen higher than that caused by the rear bearing. A cage defect frequency is calculated from a roller cage movement and it is caused by a roller size variation. Although it can not be identified in the FFT data due to its proximity to zero, the periodicity of the data in the time domain in Figure 2 indicates the existence of a modulating frequency component around 0.0325 Hz which corresponds to the modulation between the nose bearing cage frequency and the rear bearing cage frequency. This low modulation frequency requires that the number of revolutions for the sample data to evaluate the axis of rotation errors should be large enough to include at least one period of the cycle.

The effect of the spindle shaft and the housing alignment was examined next. The asynchronous radial error motion at a location about 3 inches from the spindle nose before remachining of the housing ranges between 0.9 micrometer to 1.4 micrometer up to 650 rpm spindle speed, whereas within the same speed range, the asynchronous error decreased to 0.4 to 0.5 micrometers after the housing was remachined. Similarly, the average error motion at the same location dropped from 0.64 micrometers to around 0.06 micrometers after remachining of the housing. Sample polar plots of radial error motion of the spindle at 650 rpm before and after remachining are shown in Figure 5.

DISCUSSION

This study shows us the power of axis of rotation error measurements in analyzing spindle design and bearing selections. The extremely good match between the calculated bearing frequencies and the measured spindle frequencies indicates that using the measurement data we can develop strategies to improve spindle performance by selecting proper bearings using published data. For example, it is possible to change bearing diameter, taper angle, number of rollers, or precision grade to modify error motions.

This ongoing investigation also verifies that using high-precision bearings alone does not guarantee a high-precision performance out of the spindle if the design and/or manufacturing of the spindle components are not adequate. We also have learned during the remachining of the housing that roundness of the bearing seats can significantly degrade if the housing is not supported properly or if there is differential heating and/or cooling. Therefore, in the second half of the experiments we avoided bolting the spindle to the base plate and monitored the temperatures of the spindle to minimize structural distortions.

The evaluation of this spindle will continue by measuring spindle thermal drift, dynamic and static stiffness and damping. At the end of these tests, a series of cutting tests will be carried out on the spindle to correlate the measured performance parameters to the actual cutting performance.

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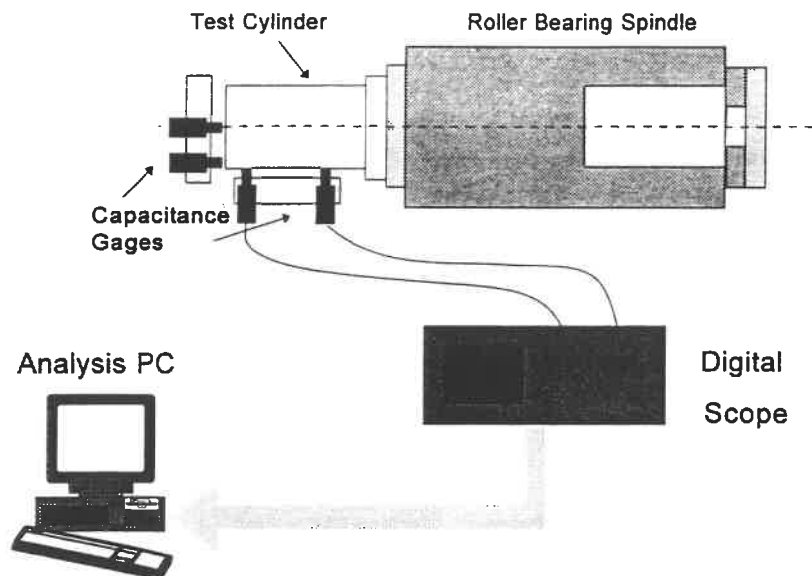


Figure 1 Axis of rotation error measurement setup

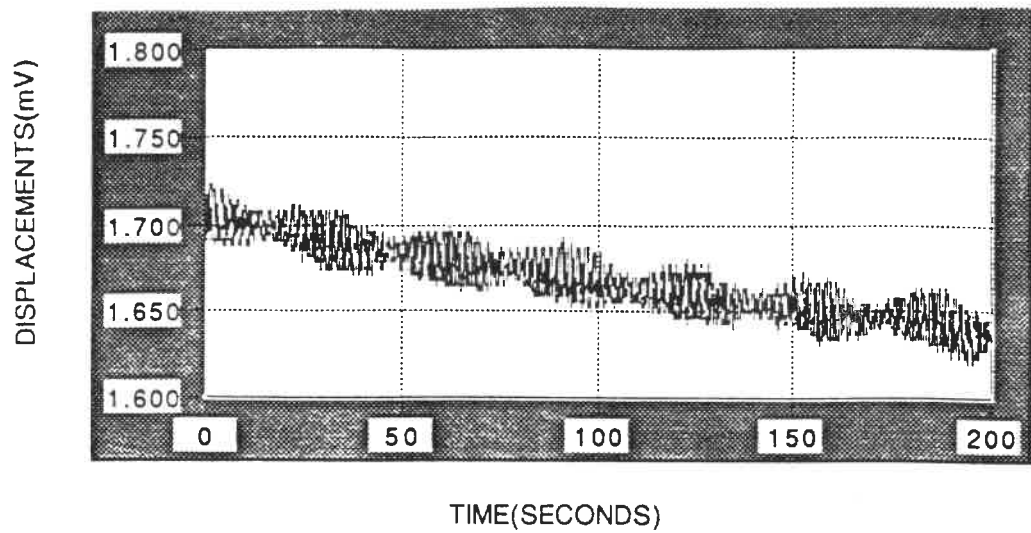


Figure 2 Radial spindle displacement measured by capacitance probe (1 sample/revolution)

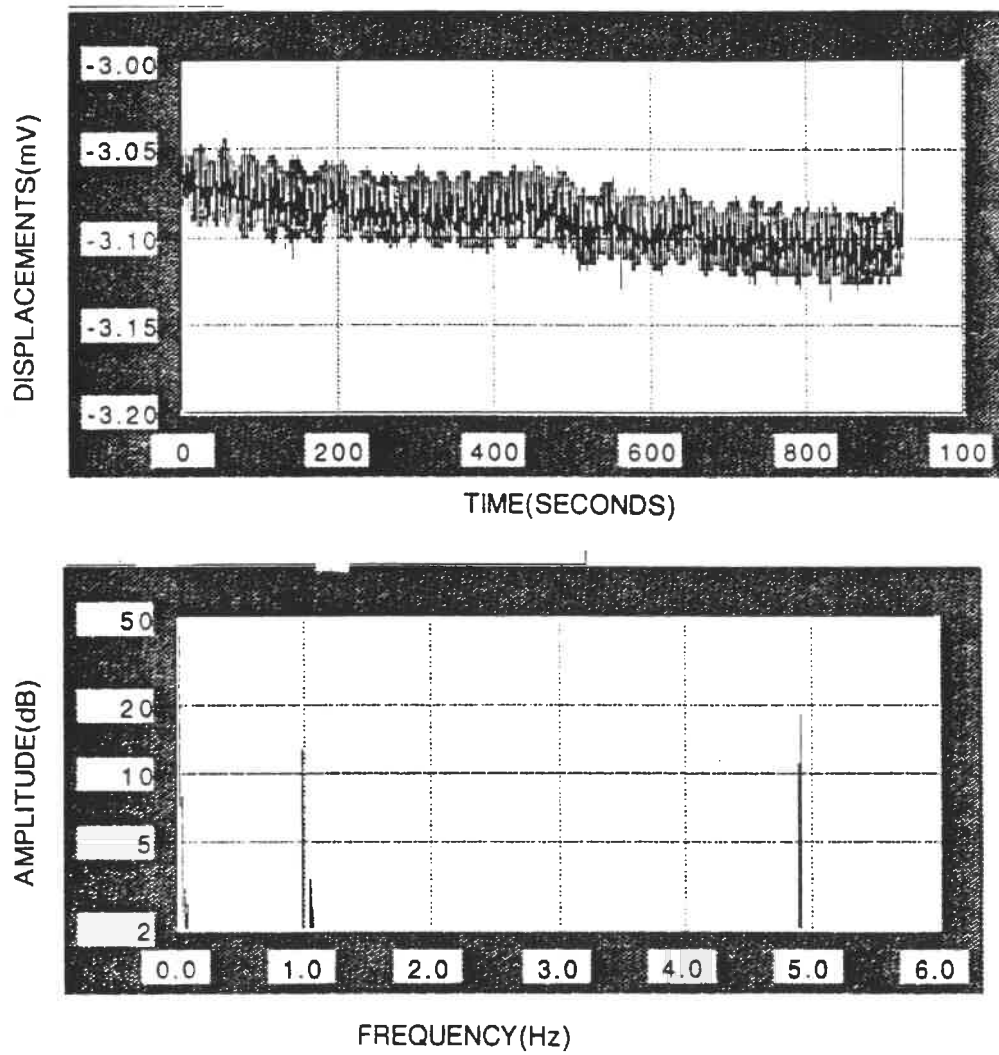


Figure 3 Spindle radial displacement in time and frequency domains

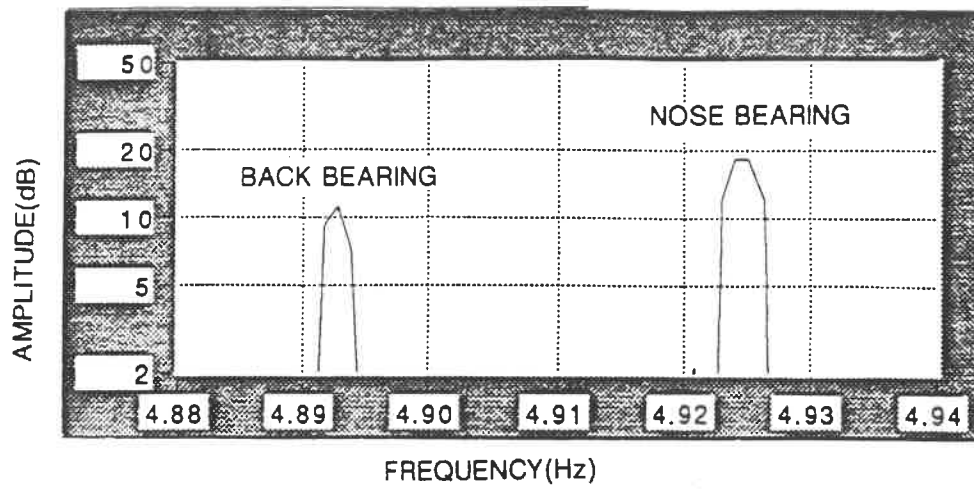


Figure 4 Bearing roller cage frequencies

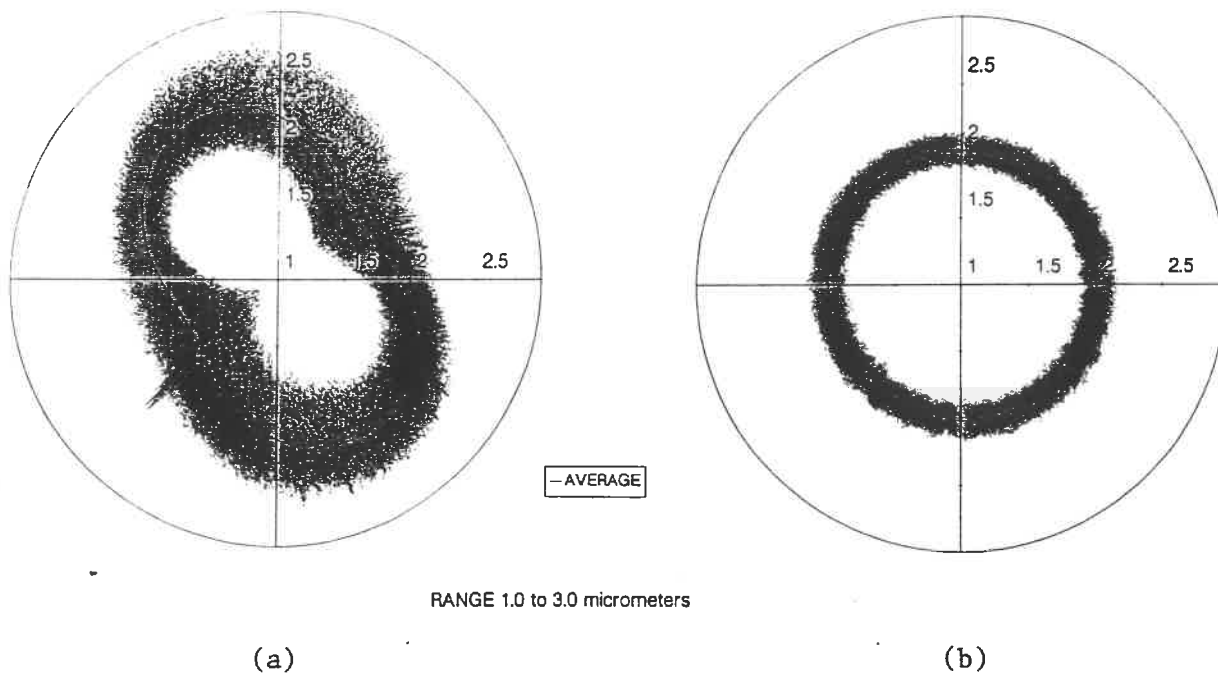


Figure 5 Radial error motion at 650 rpm, before (a) and after (b) remachining of housing

Proposal of multi-point ultraprecision slide feed mechanism by making use of force-operated actuator

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1. Introduction

In designing a slide feed mechanism of precision machine, it should be careful that a positioning accuracy at the position detecting point of the slide with feedback sensor can not be a positioning accuracy at another point of the slide in a range of feed resolution of nanometer.

Internal and external loads applied to a slide cause an elastic deformation and rotational error of the slide.

In the present paper, new principles of multi-point drive ultraprecision slide feed mechanism is proposed as a new design concept of slide feed mechanism in consideration of an elastic deformation of the slide and is experimentally extended.

2. Proposal of multi-point drive slide feed mechanism

In the conventional slide feed mechanism, one controlled object is driven with one actuator. That is, it is a system of applying a driving force to one point of the controlled object and positioning it.

An object -moving body- in positioning feed system is applied three different kinds of loads as shown in Fig. 1, an movement load such as sliding load along guideways, an inertial force and so on, a working load such as machining load of machine tools and a driving force of actuator against these loads. These three kind of loads applied to a moving body cause inevitably an elastic deformation of moving body and result not negligible motion error of precision motion mechanism.

If it is realizable that one moving body is driven at several driving points using several actuators so as to be minimized of an elastic deformation of the moving body and also a sharing ratio of driving forces of the actuators is adjusted so as to be optimum, a new design principle can be introduced to an ultra precision motion mechanism.

Actuators applied to a positioning servo system are classified into a displacement transmission actuator and a force transmission actuator according to the operation principle[1]. A positioning mechanism making use of several displacement transmission actuators such as a lead screw feed mechanism does not have a function of actively adjusting ratio of driving forces of the displacement transmission actuators in operation principle.

The authors have proposed a force operated positioning mechanism using a force transmission actuator and have developed the mechanism with high feed resolution and also high stiffness[2].

In the present paper, a positioning servo system of feeding one moving body with two force transmission actuators is constructed and it is experimentally shown that a multi-point drive ultraprecision positioning mechanism can be realized with no positional interference between the actuators.

Fig. 2 shows a block diagram of the multi-point drive positioning mechanism using an example of the mechanism with two force transmission hydraulic actuators. The system involves one controlled object, two actuators and one feedback sensor. It is possible to share the driving force of every actuator optimum against loads by adjusting gain of a servo amplifier of every actuator. The sharing ratio of the two driving forces can be continuously adjusted on-line for moving states of a controlled object.

3. Test rig and experimental results

Fig. 3 is a construction of the test rig. A controlled object is a slide with dimensions of 500^L x 300^W x 234^H and mass of 150 kg vertically supported by plain bearing guideways and horizontally by hydrostatic bearings on the frame structure. A contact pressure of plain bearings can be changed by adjusting hydrostatic pressures of coupled hydrostatic pads to the plain bearings. Forced supply of lubricating oil with viscosity grade of ISO #22 is applied to the plain bearings.

The slide is horizontally driven by two force transmission actuators respectively mounted on the both ends of frame structure and the system constructs a two-point drive positioning mechanism. A displacement of

the slide is detected at lower center of frame structure along the plain bearing guideways with a capacitive displacement sensor fixed on the structure and is fed back.

Drive coils of the two electro-hydraulic servo valves are connected in parallel and are driven by one current amplifier.

A series of experiments for several conditions is made for checking a function of controllability of sharing ratio of two driving forces of the actuators.

Fig. 4 is records of the slide displacement at the feedback point (Fig. 4a) and two driving forces of the actuators (Fig. 4b,c) for repeated reciprocations with feed rate of 100 nm/sec. in the travel range of 4 μm . Driving forces are 124N and 152N respectively and a good repeatability is shown for repeated reciprocations.

Fig. 5 shows the sharing rate of two driving forces of the actuators for several bearing pressures and feed rates in the travel range of 4 μm . The driving force ratio is controlled within 3 % deviation approximately and it is experimentally shown that a multi-point drive system with force transmission actuators has a function of controlling driving force ratio at the driving points of slide. Here, the summarized driving force for the plain bearing pressure of 90 kPa is approximately twice of one for 70 kPa.

Fig. 6 and 7 are records of the slide displacement at the feedback point and driving forces of the actuators for stepwise feed commands of 10 nm/step and 1 nm/step. The driving forces show a stepwise response similarly to the stepwise slide displacement. The slide displacement at the feedback point for the feed command of 1 nm/step is well controlled and any displacement interference between the actuators can not be recognized.

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Key Words : positioning servo mechanism, multi-point drive mechanism, force-operated positioning system, feed resolution, Abbe's offset

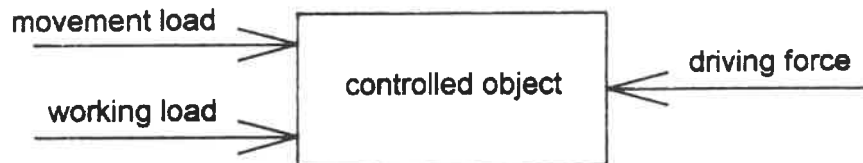


Fig.1 Forces applied to controlled object

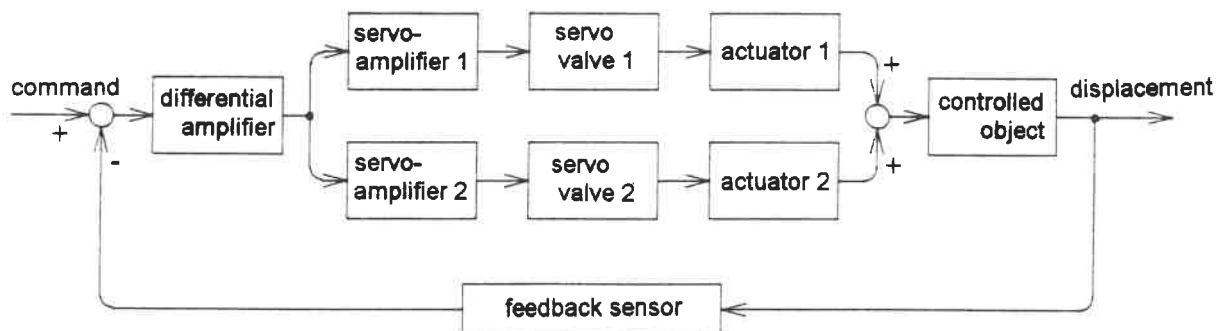


Fig.2 Block diagram of multi-point drive positioning servo system

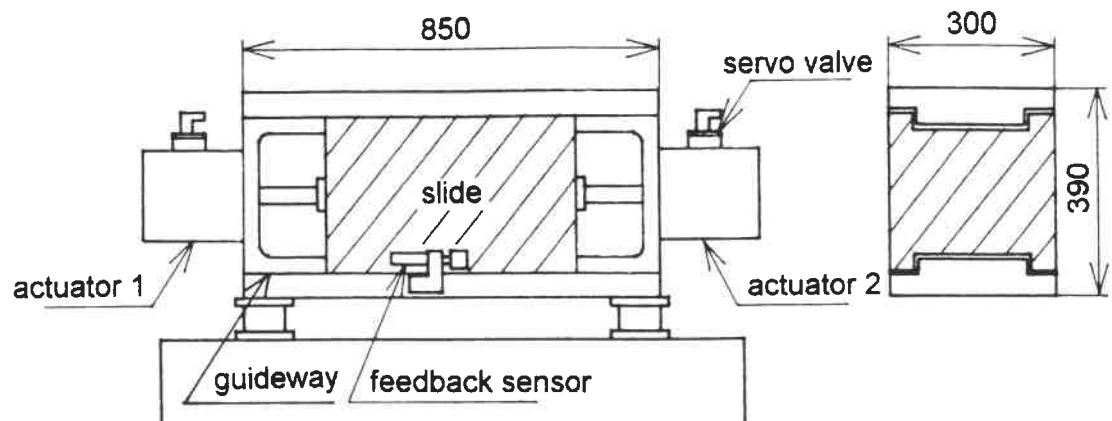
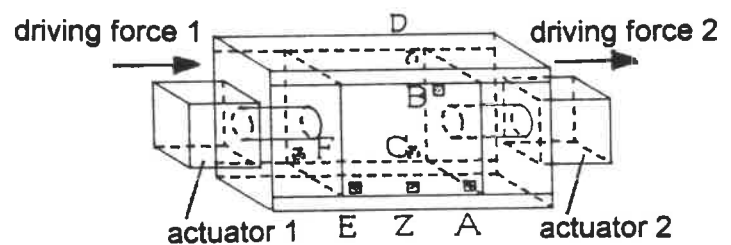
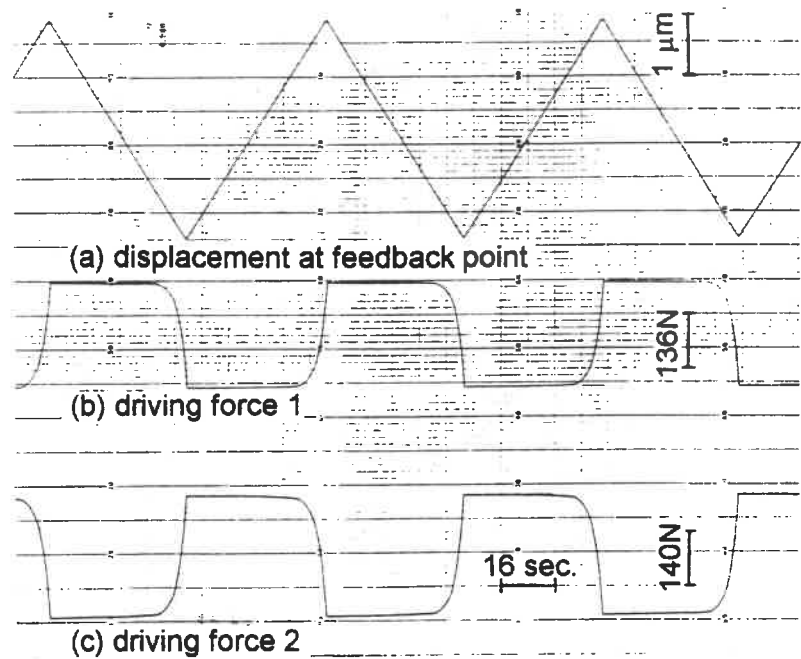


Fig.3 Test rig



A-F : displacement sensor

Z : feedback sensor

Fig.4 Reciprocation with feed rate of 100 nm/sec.

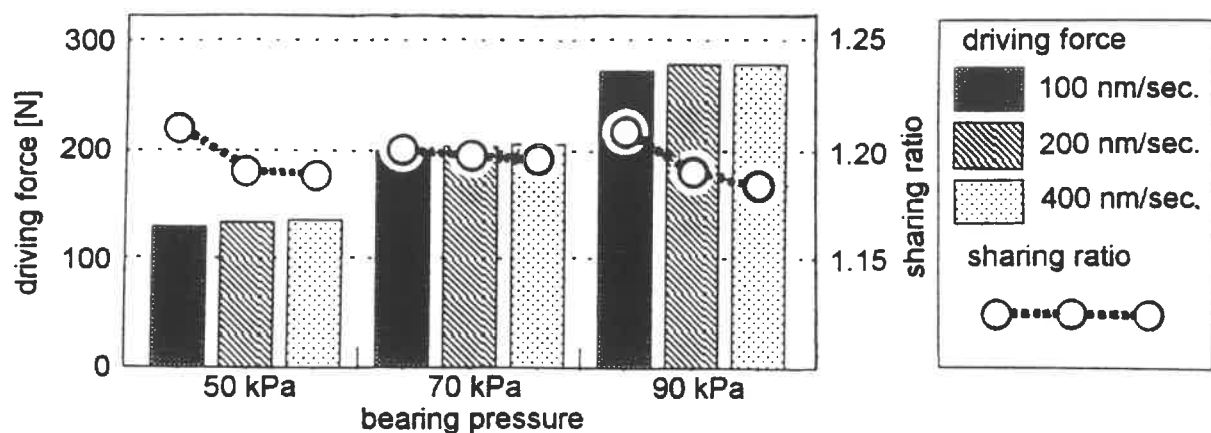


Fig.5 Sharing ratio of driving forces

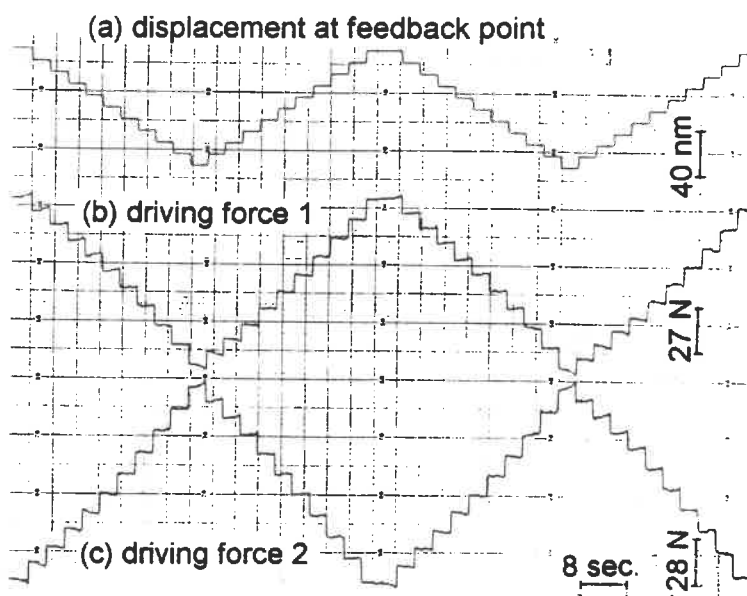


Fig.6 Stepwise feed of 10 nm/step

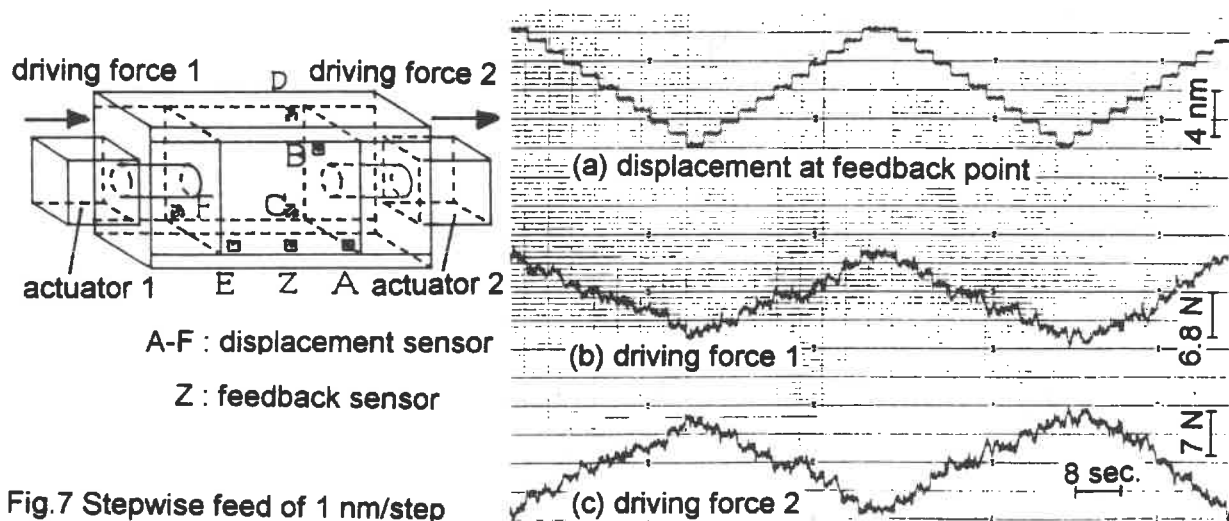


Fig.7 Stepwise feed of 1 nm/step

Design Optimization of Herring Bone Air Bearings for Polygon Mirror Laser Scanners

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1. Introduction

Recently, self-acting air bearings have been applied to computer peripherals such as, laser printers and scanners for high quality output and input. In this paper, the characteristics of the herring bone bearing shown in Fig.1 will be discussed. Among various self-acting air bearings, this type of bearing has been considered to be the most suitable for high speed applications due to its high load capacity and high stability. To optimize the bearing design, the relationships between design specifications and bearing characteristics, such as load capacity and stability, have to be known. However, there are few design examples and design data available, and they are insufficient for design optimization. Therefore, the design of herring bone bearings still depends mainly on the experiences. In this paper, the computational analysis of partially grooved herring bone bearings has been presented and the information needed for the design optimization of the bearing is obtained.

2. Nomenclature

b_g : width of a groove
 b : sum of a groove width and a land width
 e : eccentricity of bearing
 D : diameter of bearing, $D \approx 2R$
 h : bearing clearance, $H = h/h_o$
 h_o : average clearance over the land
 h_g : depth of the grooves, $H_g = h_g/h_o$
 L : length of the bearing
 l_g : length of the grooves in bearing longitudinal direction
 n : number of the grooves
 p : pressure within the clearance
 p_a : ambient pressure, $P = p/p_a$
 R : radius of the bearing
 w : load capacity of the bearing, $W = w/(p_a LD)$
 x : coordinate on bearing surface in circumferential direction, $X = x/R$
 y : coordinate on bearing surface in longitudinal direction, $Y = y/L$
 Z : normalized groove length, $Z = 2 l_g/L$
 α : width ratio of the grooves, $\alpha = b_g/b$
 β : pump-in angle of the grooves

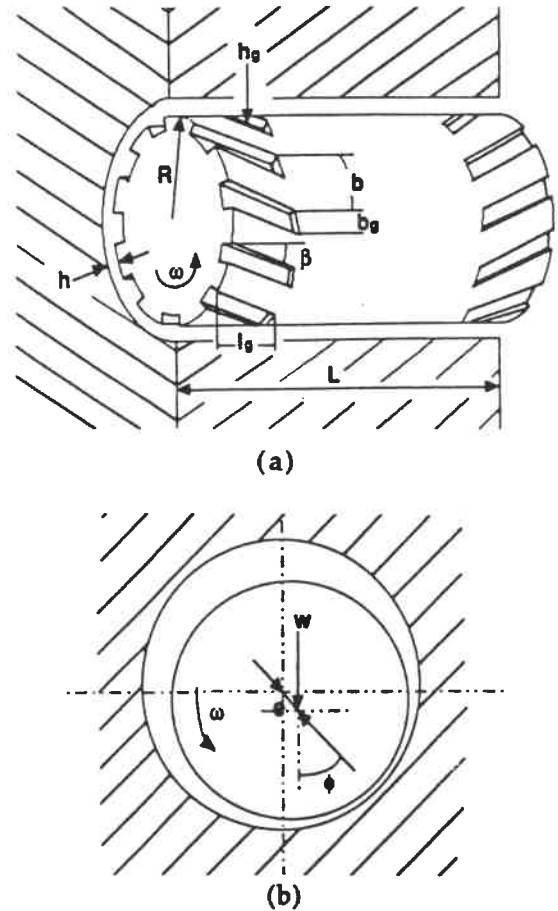


Fig. 1 Configuration of herring bone bearing

ε : eccentricity ratio, $\varepsilon = e/h_o$
 ϕ : attitude angle
 A : bearing number, $A = 6\mu\omega(R/h_o)^2/p_a$
 μ : viscosity of air
 ω : rotational speed

3. Method of analysis

The characteristics of the air film within the bearing clearance can be described by a normalized Reynolds equation:

$$\frac{\partial}{\partial X}(H^3 P \frac{\partial P}{\partial X}) + \frac{\partial}{\partial Y}(H^3 P \frac{\partial P}{\partial Y}) = A \frac{\partial(PH)}{\partial X}$$

Generally, the strict solution of this equation is not possible. So the FEM is employed to obtain a

numerical solution. Once the solution, i.e., pressure distribution over the bearing surface is obtained, the bearing performance such as load capacity and attitude angle can be derived.

4. Validation of the computational results

In order to confirm the validity of the programs developed in this study, a comparison of the computed results with the experimental data from Malanoski[2] as shown in Fig.2 has been done. It is seen that a reasonably good agreement has been obtained, but the divergence between the experiments and computations increases with the increase in the bearing speeds.

Though it is not shown in this paper, a comparison between the data from the finite element computation and the data from the Castelli and Vohr's method[3], which assumes the bearing have infinite number of grooves, has been done. According to the comparison, the results from the two methods are often close, but the finite element analysis agrees better with the experiments in low speeds.

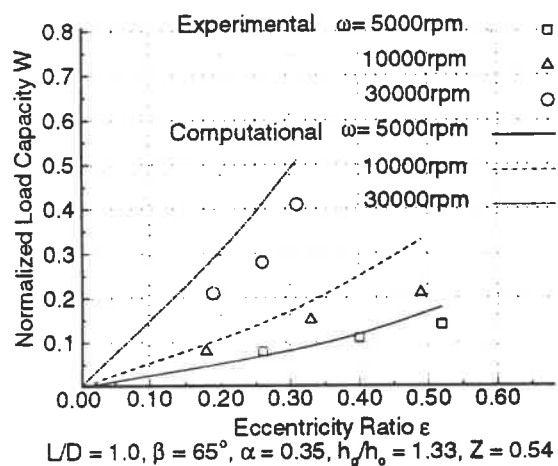


Fig.2 Comparison with experimental data

In addition, convergence checks have been done in this study because the computational results may vary significantly with the meshes upon which the computation is carried out. In this paper, all the results are obtained with reasonably large number of elements, and the divergence from the computational results obtained with higher number of elements is less than 5%.

5. Bearing loading performance

The performance of a herring bone bearing is mainly determined by design specification (see figure 1 and nomenclature), such as, the groove

depth h_g ; the groove width b_g ; the pump in angle β and the groove length Z .

The most commonly used characteristics for a bearing performance are normalized load carrying capacity W (see nomenclature) and attitude angle ϕ (see fig. 1). W represents a loading capacity per unit area ratio and ϕ are used to give indirectly the stability of the bearing (i.e., bearings with a large attitude angle ϕ are more liable to become instable[4]). The rotation of the spindle in all the discussions below is in the direction that pumps air into the bearing clearance.

Figure 3, 4 and 5 are the relationships between the pump-in angle β and the bearing performance obtained by the computations at $L/D=0.75$, 1.0 and 1.5. Figure 3 shows that increasing pump-in angle β can increase the load capacity W . The optimal value of ϕ is obtained around $\beta=55^\circ$ with respect to the normalized groove depth $h_g/h_o = 1.2$.

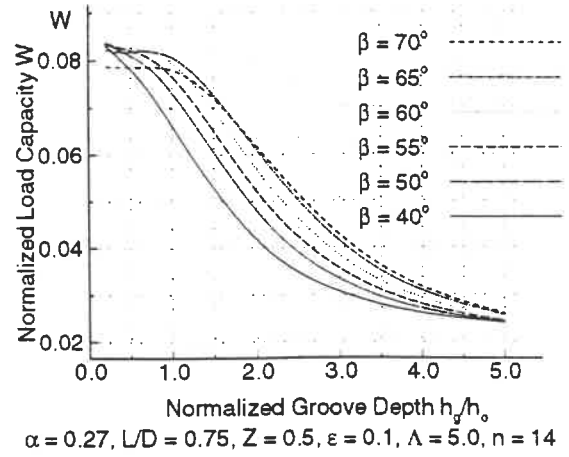
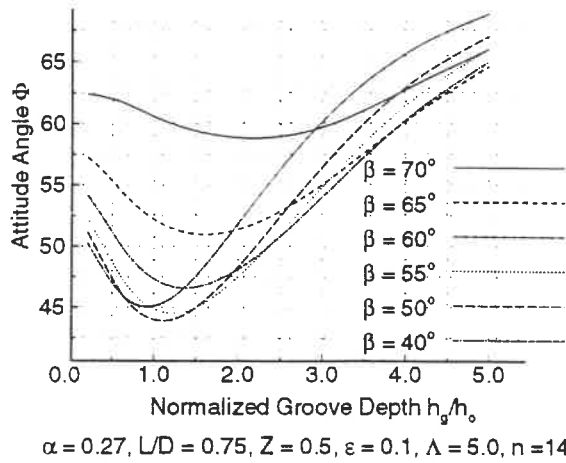
Figure 4 shows the relationship between β and the performance of the bearings with $L/D=1.0$. The trend in this figure is similar to that of the bearings with $L/D=0.75$. But the optimal values for the bearings of $L/D=1.0$ are obtained around $\beta=50^\circ$ with $h_g/h_o=0.8$, which are smaller than the ones obtained with $L/D=0.75$. In addition, some readers may have noted that these optimal values are considerably smaller than the commonly used ones offered by Castelli and Vohr[3] which is $\beta=65^\circ$ and $h_g/h_o=1.33$.

Figure 5 shows the relationship between β and the bearing performance when $L/D=1.5$. Apparently, the optimal value of β for the minimum attitude angle only exists when it is smaller than 50° . The optimal values for overall performance may be obtained at about $\beta=40^\circ$ with $h_g/h_o=0.5$.

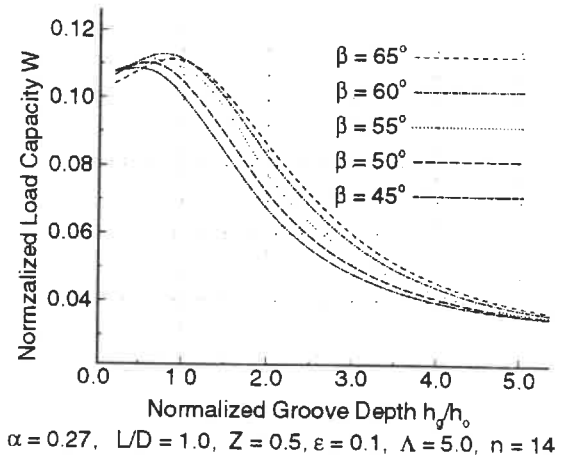
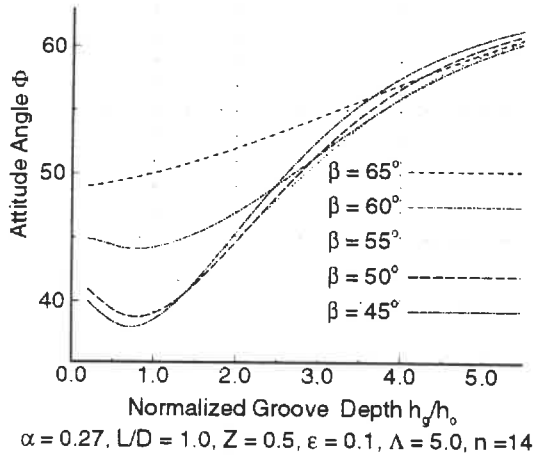
Combining the results in figures 3, 4 and 5, it is known that a bearing with an larger L/D ratio will have a smaller attitude angle. But it should be designed with smaller β and h_g/h than those used in bearings with smaller L/D ratios. However, readers should note that the instability caused by factors other than the attitude angle ϕ increases with the increase in the L/D ratio.

The effect of the groove width α has been shown in Figure 6. According to the computed results, decreasing the groove width α can increase the bearing load capacity W , but also, increases slightly the attitude angle ϕ .

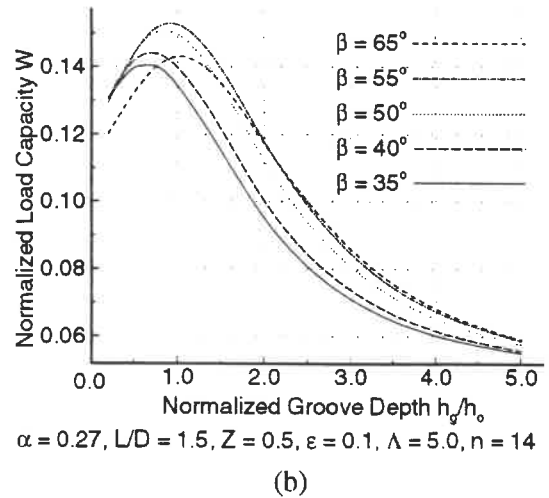
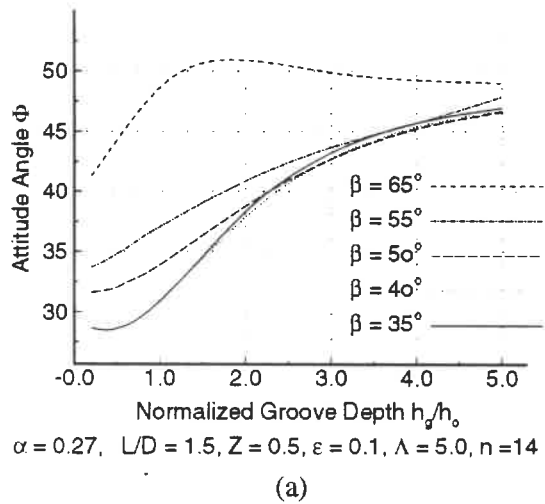
The influence of the groove number n is shown in Figure 7. Apparently, the load capacity W increases while the attitude angle ϕ decreases with



(a) (b)
 Fig. 3 Influence of the groove pump-in angle β , $L/D = 0.75$



(a) (b)
 Fig. 4 Influence of the groove pump-in angle β , $L/D = 1.0$



(a) (b)
 Fig. 5 Influence of the groove pump-in angle β , $L/D = 1.5$

the increase of the groove number n . But the change caused by the number of the grooves is relatively small.

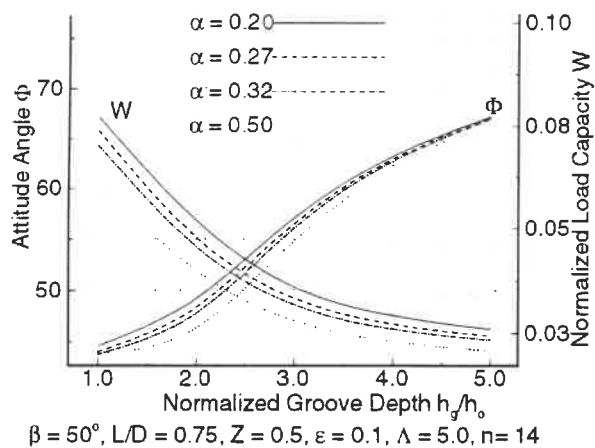


Fig. 6 Influence of the groove width α

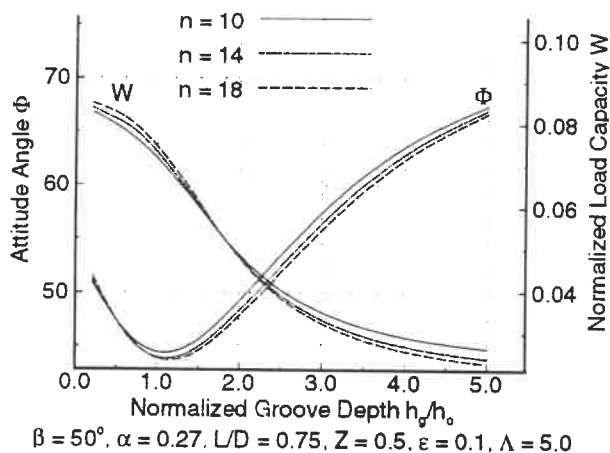


Fig. 7 Influence of the groove number n

Figure 8 shows the relationship of the eccentricity ratio of the bearing ε to the bearing performance. Naturally the load capacity W increases with an increase in the eccentricity ratio ε but the attitude angle ϕ decreases.

Figure 9 shows that bearing load capacity W increases approximately in proportion to the bearing number Λ , while the attitude angle ϕ decreases. However, the reader should notice that the instability caused by factors other than the attitude angle ϕ increases with the increase in the bearing number Λ .

6. Conclusion

Based on the computed results and foregoing discussions, the influences of the design parameters

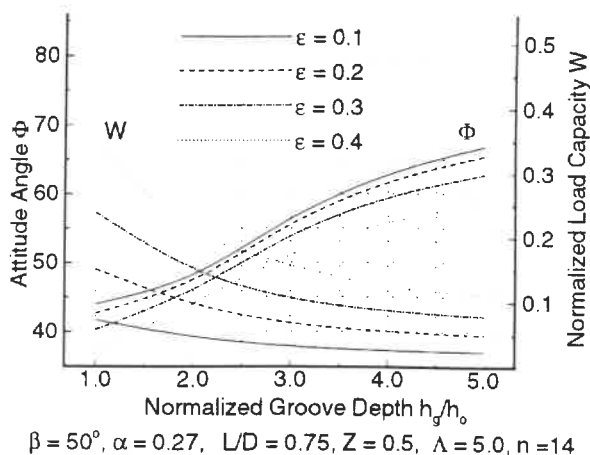


Fig. 8 Influence of the eccentricity ratio ε

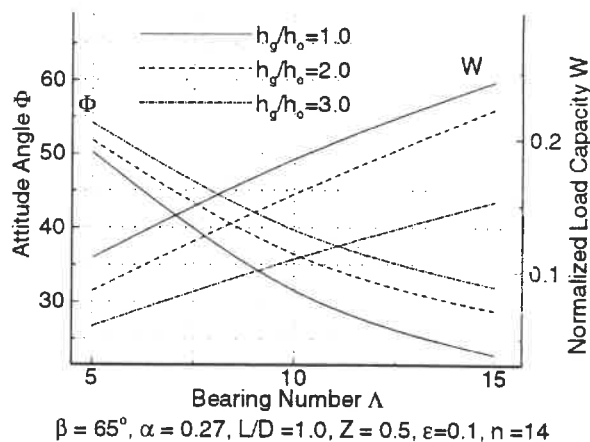


Fig. 9 Influence of the bearing number Λ

on the load capacity W and the attitude angle ϕ have been known. Therefore, the design optimization of the partially grooved air bearings becomes possible.

7. Acknowledgment

The authors would like to express their appreciation to the National Center for Supercomputing Applications.

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An Active Inherent Restrictor for an Infinite-Stiffness Aerostatic Bearing

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1. Introduction

For improving the stiffness and the rotational accuracy of aerostatic bearings, many active control methods have been proposed. These methods can be divided into two groups by the energy source for controlling; one group uses the pneumatic pressure on the bearing surface to control the opening of the restrictor^{1),2)}, and another group uses external electric power for controlling^{3),4)}. In the present paper, we propose an Active Inherent Restrictor (abbreviated as "the AIR") which belongs to the latter group. The restriction region of the AIR is placed on the bearing surface, and the opening of the restrictor (air film gap) can be changed by a piezoelectric actuator. When some displacement of the shaft is detected by a sensor, a micro computer controls the piezoelectric actuator so as to eliminate the shaft displacement. Thus high stiffness and rotational accuracy can be obtained. The AIR is applicable to the aerostatic bearings of all types which use conventional passive inherent restrictors. In the present paper, the AIR is incorporated into an aerostatic thrust bearing to analyze the bearing stiffness.

2. Theoretical consideration for the AIR

A schematic view of an annular pad thrust bearing with inherent restrictors is shown in Fig. 1. Outer and inner diameters of the bearing pad are D_o and D_i , respectively. Inherent restrictors are placed on the bearing surface with pitch circular diameter of D_r . The number of the restrictors is n and the diameter of the inlet hole is d . As shown in the magnified view around the restrictor, air film thickness on the bearing surface is h , and air gap at the restrictor is hd . When the supply pressure is P_s , and pressure at the inlet of the restrictor is P_r , total inflow rate of the air Q_{in} is given as follows:

$$Q_{in} = \pi \times d \times h_d \times n \times c_d \times \frac{P_s}{\sqrt{R \times T}} \times \sqrt{\frac{2 \times \kappa}{\kappa - 1}} \times \sqrt{\left(\frac{P_r}{P_s}\right)^{\frac{2}{\kappa}} - \left(\frac{P_r}{P_s}\right)^{\frac{\kappa+1}{\kappa}}} \quad (1)$$

where R : gas constant, T : absolute temperature, C_d : flow coefficient and k : the adiabatic exponent. This inflow air is exhausted from the inner and the outer periphery of the bearing surface. Total outflow rate Q_{out} is given by equation (2),

$$Q_{out} = \frac{\pi \times h^3}{12 \times \mu \times R \times T} \times \left\{ \frac{1}{\ln(D_o / D_r)} + \frac{1}{\ln(D_r / D_i)} \right\} \times (P_r^2 - P_a^2) \times C_f \quad (2)$$

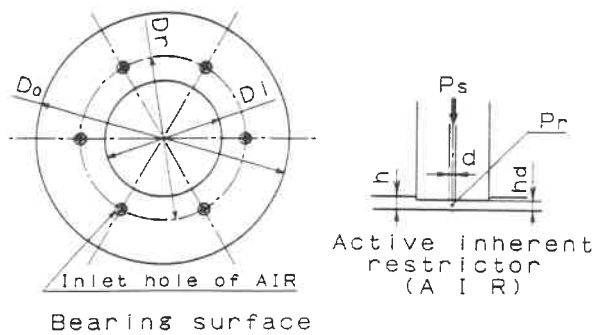


Fig. 1 An annular pad thrust bearing with inherent restrictors

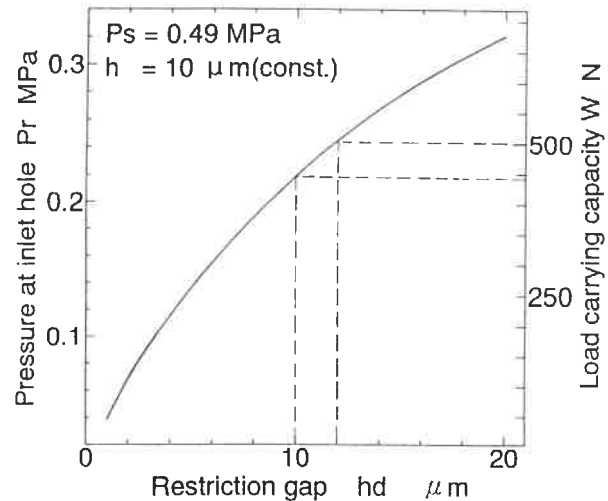


Fig. 2 Pressure on bearing surface controlled by restriction gap

where, P_a : ambient pressure, μ : viscosity of the air and C_f : correction factor. By equating Q_{in} and Q_{out} , P_r can be calculated. Then the load carrying capacity W of the bearing can be obtained by the product of pressure P_r and the virtual bearing area S_e .

When the applied load increases by dW , P_r should increase for supporting this additional load. The restriction gap of a conventional passive inherent restrictor is always equal to the air film thickness on the bearing surface. Therefore, to increase the pressure P_r with such a conventional restrictor, air film thickness h should decrease by dh (the restriction gap also decreases by dh), and the finite bearing stiffness K is given by $K = dW/dh$. On the other hand, the restriction gap of the proposed Active Inherent Restrictor (the AIR) can be increased for increasing the bearing pressure P_r , while the air film thickness h is kept constant. For constant air film thickness h , the relationship between the restriction gap hd and the bearing pressure P_r is calculated from eqs. (1) and (2), and shown in Fig. 2, where the specification of the calculated bearing follows that of the thrust bearing shown in the next section. Figure 2 indicates that P_r can be increased by increasing hd , and the additional load can be supported without changing the air film thickness h . Consequently, the bearing stiffness is infinite.

Most of active restrictors which have been proposed are not used in practice because of the occurrence of pneumatic hammer and the difficulties in designing, manufacturing and assembling. The AIR is basically an inherent restrictor, therefore the danger of the occurrence of pneumatic hammer is little, and it is possible to incorporate the AIR into most of the aerostatic bearings instead of conventional inherent restrictors.

3. A thrust bearing with the AIR

Figure 3 shows an air-spindle which is composed of an opposed-pad aerostatic thrust bearing with the AIR (outer diameter=65mm and inner diameter=30mm), and an ordinary aerostatic radial bearing (inner diameter=29mm and length=50mm). On each thrust bearing surface, there are six inherent restrictors. Among these restrictors, three restrictors on the top bearing surface are the AIR. Cross sectional view in Fig. 3 shows the incorporated AIR and the capacitance transducer for detecting the shaft

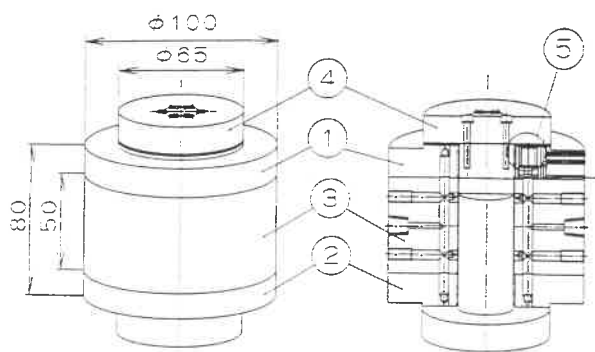


Fig. 3 Air-spindle with the AIR
1;thrust bearing with the AIR,
2;thrust bearing, 3;radial bearing,
4;thrust plate of shaft, 5;the AIR

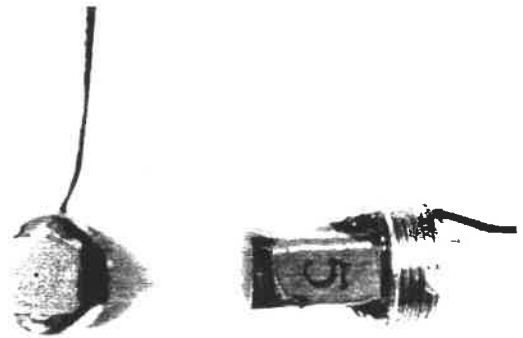


Fig. 4 Piezoelectric actuator used in the AIR

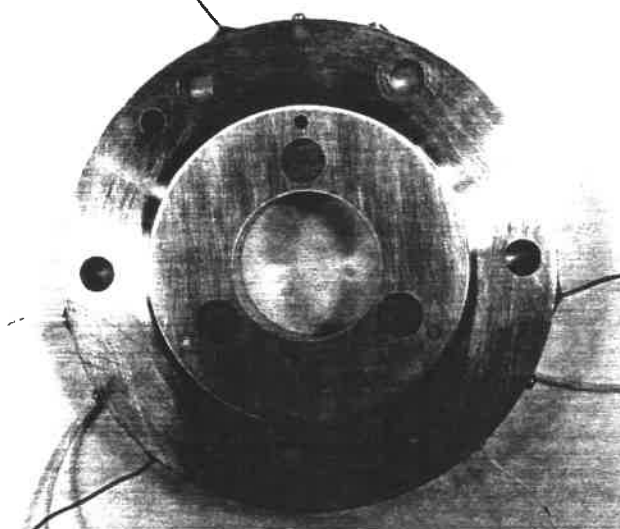


Fig. 5 Thrust bearing surface with the AIR

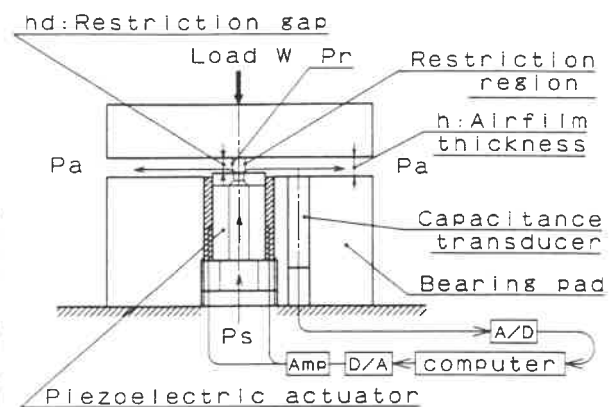


Fig. 6 Block diagram of control system

displacement. The radial bearing has two-rows of restrictors, and each row has six passive inherent restrictors. The air film thickness on the thrust bearing surface is $10\mu\text{m}$, and that of on the radial bearing surface is $20\mu\text{m}$.

The piezoelectric actuator used as the AIR is shown in Fig. 4; the actuator is 5mm square and 10mm length, and it has a longitudinal through hole of 1mm diameter. Diameter of one end of the through hole is reduced to be 0.3mm to function as an inherent restrictor. The screw on the other end of the actuator is used for incorporating. The piezoelectric actuator expands about $8\mu\text{m}$ for supply voltage of 150V. The thrust bearing surface where the AIRs are incorporated is shown in Fig. 5. There are six inlet holes; three holes in black square parts are the AIRs, while another three are ordinary inherent restrictors. Silicon rubber is filled in the gap between the AIR and the bearing surface. In the neighborhood of each piezoelectric actuator, the tip of a capacitance transducer can be seen. Resolution of this capacitance transducer is 10nm.

converted and inputted to the micro-computer. To eliminate the shaft displacement, the computer calculates the supply voltage to the piezoelectric actuator by using PI control algorithm, and the calculated voltage is outputted to the driving amplifier via D/A converter.

4. Load carrying capacity

Thrust load carrying capacity of the air spindle shown in Fig. 3 was analyzed. Weights were loaded on the upper thrust plate, and the displacement of the plate was measured by electric micrometers.

Figure 7 shows the measured displacement and the voltage for the piezoelectric actuators. When the AIR is not used (without AIR), the thrust bearing is an ordinary bearing with passive restrictors, and the displacement increases with the increase of the load. Therefore, a finite stiffness of $40\text{N}/\mu\text{m}$ was obtained. When the AIR is used (with AIR), the displacement with the increase of load is little while the voltage for the piezoelectric actuator decreases so as to increase the restriction gap and the pressure on the bearing surface. Therefore, it is shown in Fig. 7 that in a load range lower than 60N, the AIR is effective and the bearing stiffness can be infinite. During these experiments, pneumatic hammer did not occur.

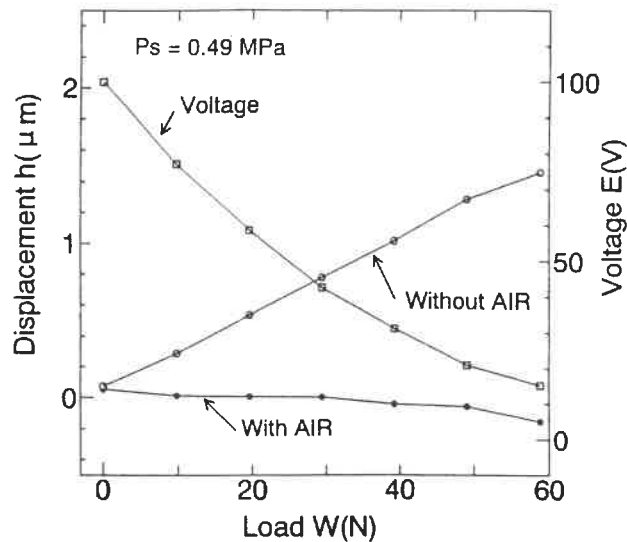


Fig. 7 Thrust load carrying capacity

5. Summary

The proposed Active Inherent Restrictor (AIR) is incorporated into an aerostatic thrust bearing, and the effect of the AIR on the bearing stiffness is analyzed. From the analysis, it is shown that an infinite bearing stiffness can be obtained for the thrust load less than 60N. Pneumatic hammer didn't occur during whole experiments. Therefore, the AIR is effective for improving the performance of aerostatic bearings.

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Residual Error Compensation of a Vision-based Coordinate Measuring Machine¹

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Introduction

The commercial equipment for measuring lithography patterns has reached an accuracy level which makes it difficult to provide two dimensional artifacts of higher accuracy for use as calibration standards. Because of the simpler nature of one-dimensional measurements, it is still possible to measure one line of a grid plate to a higher accuracy than is possible using the two dimensional measurement machines. The main idea of this paper is to introduce the idea of using a partially calibrated grid plate (one or two lines calibrated) and a series of measurements of the whole plate on the two dimensional machine to produce a calibration of both the grid plate and the machine. Our procedure involves measurement of a grid plate in three specified orientation. The measurement data is used to build an error compensation model. The following sections give a brief description of the method.

Software compensation of a coordinate measuring machine has become an accepted practice to enhance the accuracy of a CMM. In addition to parametric error compensation schemes, research work in self calibration schemes have also been carried out in recent years. Hocken and Borchardt used the measurements of an artifact in two specified orientations to model squareness and scale errors[1]. Belforte, et al. developed a model that included the twenty one error components. They used Legendre polynomials to model the errors[2]. The self calibration methods eliminate the need for calibrated artifacts, but require the dimensional stability of the artifacts during the measurement. Raugh has also developed a method that falls under this category[3]. Our method is an extension of the Hocken-Borchardt method.

Mathematical Model and Validation

The first measurement of the grid plate is done after aligning one of its lines parallel to one of the machine axis. In this position the coordinates (xg_i, yg_i) of the grid points, in the grid plate coordinate system can be related to the coordinates of the grid points in

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the machine coordinate system $(x_{m_{ij}}, y_{m_{ij}})$ by a transformation matrix representing the machine errors as shown below.

$$\begin{matrix} x_{g_i} \\ y_{g_i} \end{matrix} = T_{m_{ij}} \begin{matrix} x_{m_{ij}} \\ y_{m_{ij}} \end{matrix} \quad \begin{matrix} i = 1, 2, 3 \\ j = 1, 2, \dots, N \end{matrix}$$

$$T_{m_{ij}} = \begin{pmatrix} 1 - \delta_x(x)_{ij} & \eta(y)_{ij} \\ (-\alpha + \epsilon_z(y)_{ij} + \delta_y(x)_{ij}) & 1 - \delta_y(y)_{ij} \end{pmatrix}$$

Where

- N = Number of grid points on the grid plate
 i = position of the grid plate ($i = 1, 2$ or 3)
 j = grid point number
 (x_{g_i}, y_{g_i}) = X and Y coordinates of the j th grid point in the grid point coordinate system
 $(x_{m_{ij}}, y_{m_{ij}})$ = X and Y coordinates of the j th grid point in the i th position of the grid plate in the machine coordinate system

The matrix $T_{m_{ij}}$ represents the effect of the parametric errors of the coordinate measuring machine. The terms

$$\delta_x(x)_{ij} = (A_x + B_x X) \quad X = x_{m_{ij}}$$

$$\delta_y(y)_{ij} = (A_y + B_y Y) \quad Y = y_{m_{ij}}$$

represent the scale error in the X and Y direction, expressed as a first order polynomial. The term

$$\eta(y)_{ij} = (\epsilon_z(y)_{ij} + \delta_x(y)_{ij}) = (A_{xy}Y + B_{xy}Y^2)$$

represents the effect of the X straightness while moving in the Y direction and the effect of the Y yaw in the X direction. The term

$$\delta_y(x)_{ij} = (A_{yx}Y + B_{yx}Y^2)$$

represents the effect of Y straightness while moving in the X direction.

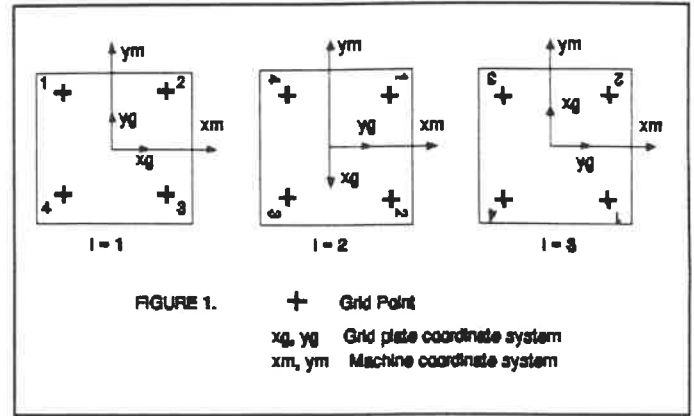
$$\epsilon_z(y)_{ij} = A_eY + B_eY^2$$

represents the Y yaw while moving in the Y direction. The out of squareness between X and Y is represented by α .

Figure 1 shows a grid plate and the two coordinate systems. In the first position the grid plate coordinates can be expressed as

$$\begin{matrix} xg_i \\ yg_i \end{matrix} = \begin{matrix} Tm_{1j} \\ Tm_{1j} \end{matrix} \begin{matrix} xm_{ij} \\ ym_{ij} \end{matrix}$$

The grid plate is then rotated through 90 degrees in the clockwise direction with respect to the Z axis. Figure 1 shows the coordinate systems and the positions of the grid points after the rotation. The grid plate coordinates in this position are given by



$$\begin{matrix} xg_i \\ yg_i \end{matrix} = \begin{matrix} xm_{ij} \\ ym_{ij} \end{matrix} R_1^{-1} Tm_{2j}$$

where R_1 is a rotation matrix.

The grid plate is measured again in a third orientation after rotating it through 180 degrees from the first position with respect to the X axis (flipped over), Figure 1. The grid plate is transparent and the grids are visible through the plate for measurement. The grid plate coordinates in this position are given by

$$\begin{matrix} xg_i \\ yg_i \end{matrix} = \begin{matrix} xm_{ij} \\ ym_{ij} \end{matrix} R_2^{-1} Tm_{3j}$$

In addition to these measurements, one line on the grid plate is assumed to be known. Let K be the distance between the points p and q on the grid plate. Then

$$xg_p - xg_q = K$$

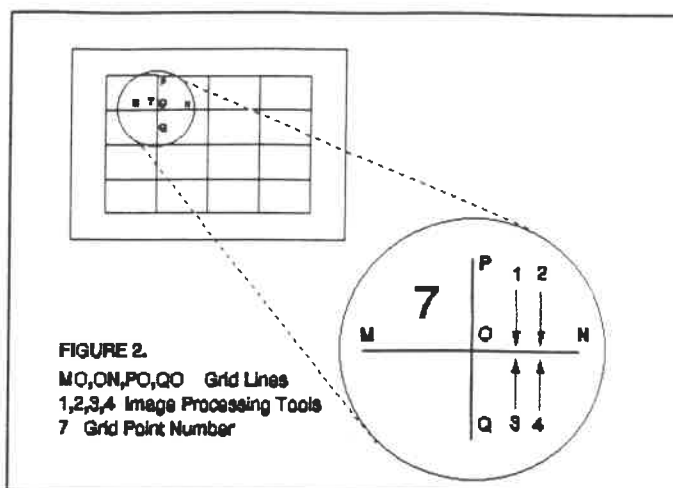
The results from these redundant measurements are then used to obtain the coefficients of the polynomial using the least squares method. The model was validated by computer simulation.

Experimental verification and conclusion

The experimental verification of the error compensation scheme was done by measuring a 584 mm x 432 mm grid plate shown in figure 2. The grid plate was measured in the three orientations described in the mathematical model. The measurement of grid point coordinates was done by first detecting the edge ON from both sides. Then a mean

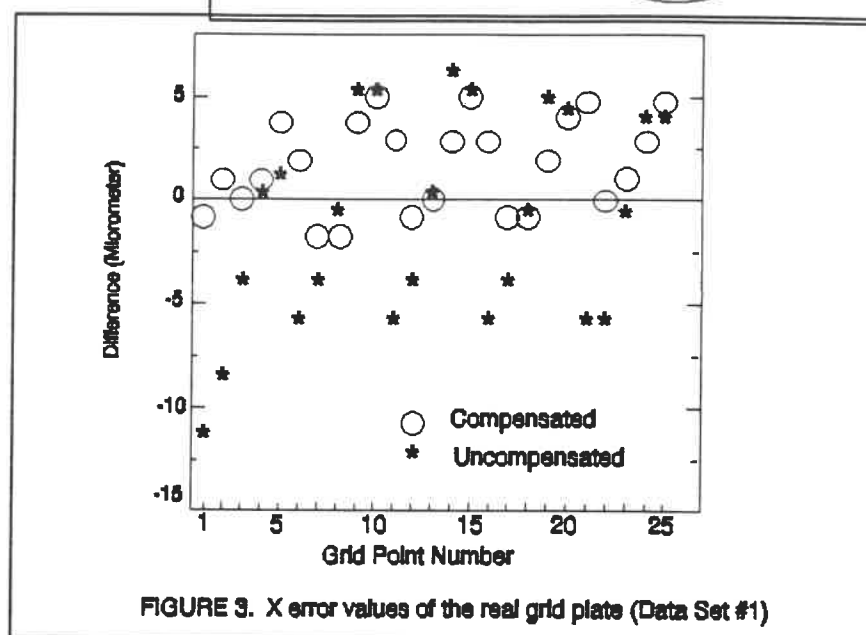
line is fit to this data.

The procedure is repeated for the lines MO, PO and QO. The intersection of these four lines give the coordinates of the grid point 7. The measured coordinates are used to compute the coefficients of the polynomials. The grid points are then compensated for the residual systematic errors.



The results for the X coordinates of the grid points, given in figure 3 shows the improvement obtained through this compensation scheme.

A self calibration method has been developed and validated using computer simulation. The model was then verified using a calibrate grid plate. The compensation scheme described in this paper can be applied in lithography machines used in the semiconductor industry.



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SAMPLING METHODS AND SUBSTITUTE GEOMETRY ALGORITHMS FOR MEASURING CYLINDERS IN COORDINATE MEASURING MACHINES

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1.0 Introduction

A coordinate measuring machine (CMM) measures discrete points on the part surfaces and computes substitute geometry to the measured set of points. This is later compared with the design specifications for tolerance verification. The accuracy of the machine and the probing system, the systematic and random deviations of the measured surface from the ideal, and the number and distribution of points measured (sampling methods) contribute to the measurement errors. The choice of algorithms used in the substitute geometry computation, their sensitivity, and bias to the input data contribute to the errors on the software side. The British Standard BS7172 gives recommendations regarding the number and distribution of points for most geometric elements [1]. A study on the behavior of the different sampling methods, in the presence of random and systematic errors for measuring circles has been done by Odyappan [2]. This paper presents the results of an investigation of the influence of the form error, sampling methods, and substitute geometry algorithms on measuring cylindrical features in CMMs.

2.0 Substitute Geometry Algorithms and Sampling Methods

The geometric element should be properly parameterized in order to compute the substitute geometry. The parameterization suggested in BS7172 is used in this study. Small changes in the geometric element, when using this parameterization, usually results in correspondingly small changes in parameter values. In this study the algorithm suggested by Forbes [3] is used for computing the least squares reference and Anderson-Osborne-Watson (AOW) [4] algorithm is used for computing the minimum zone reference. The minimum circumscribed and maximum inscribed problem are formulated as a minimum zone problem and are solved using the AOW algorithm. Various sampling schemes have been investigated and the results reported in this paper are obtained using equidistant sampling scheme. Equidistant sampling is where the distribution of points is such that the angular distance between successive points are equal.

3.0 Form Errors

In cylindrical parts form errors are present in both axial and radial (lobing) directions. An extensive literature search and survey by means of a form error questionnaire has been done by the Precision Engineering group [5]. The results of the survey revealed that for most manufacturing processes the dominant lobing is of low order

(2-7) and the lobe amplitude drops dramatically with the number of undulations per revolution. Also the possible axial form errors identified are barrel, hourglass, banana, undulated and bellmouth.

4.0 Computer Simulation

A computer simulation of the actual measurement practice was done to study the influence of form errors and substitute geometry algorithms on the different sampling methods. The simulation begins by generating a perfect cylinder. Then form errors are added on both radial and axial directions. Random noise from normal distribution is added to the coordinates to represent measurement uncertainty. Next the generated feature is sampled using different sampling techniques. Finally the sampled data is used to fit a cylinder using the four substitute geometry algorithms. The simulation is repeated 100 times for a specific sample by changing the starting point. Sample sizes from 15 to 300 points were studied. The results from the simulation were displayed in three graphs, one for the radius, one for the form error detected ($R_{max} - R_{min}$) and one for the orientation. Orientation of the axis is represented by the radius of a smallest cylinder that encloses the axis. The results (Radius) for a barrel with three lobes fitted using least squares algorithm is given in figure 2. The sampling is equidistant in three levels. For a specific number of points, the total range of the parameter values are contained within the horizontal bars. The least squares radius on a barrel or hourglass depends on the number of points at each level. Figure 3. shows the variation in radius for three different distribution of points.

5.0 Conclusion

Extensive computer simulation has been done to simulate different measurement conditions and sampling methods. Based on the results of our computer simulations, it is clear that one needs to have information regarding the manufacturing process (form error) and measurement uncertainty and the tolerance on the part, in order to choose the proper sampling scheme. However, it was also possible to arrive at some general conclusions regarding sampling schemes. Equidistant sampling at three levels is better compared to other methods such as spiral sampling. If axis straightness is desired the number of levels can be increased. The radial form error (lobing) could be detected by measuring 13, 17, 19, points per level. The tolerance level of the part and estimate of the expected lobe amplitude can be used to select the actual number of points. The least squares radius is a function of the number of points at each level. The presence of noise increases the uncertainty and the effect remains the same both in the presence or in the absence of form error .

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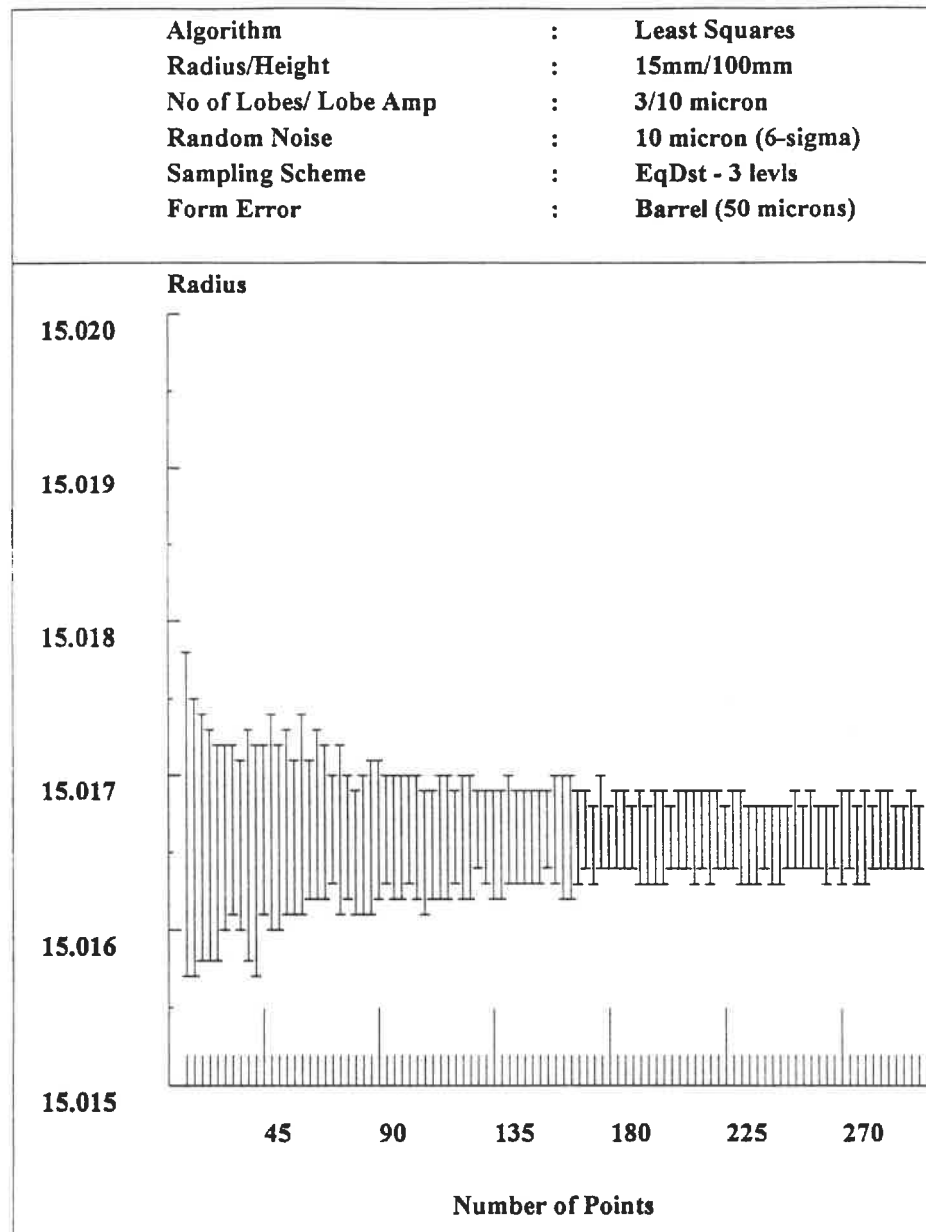


Figure 1. Results from a computer simulation for radius of a barrel shaped cylinder.

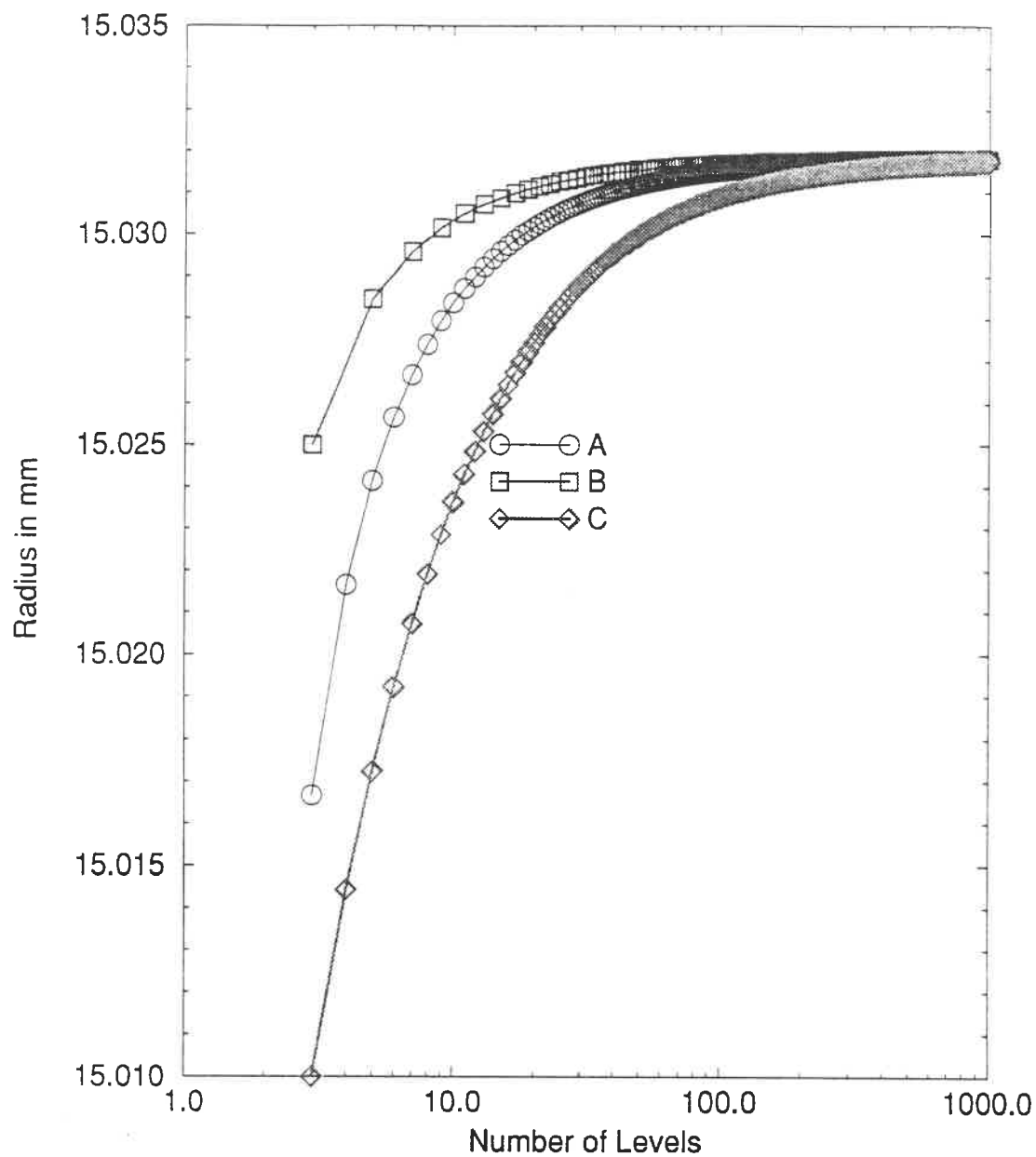


Figure 2. Variation in the least squares radius of a barrel shaped cylinder as a function of number of levels and number of points per level.

- A - Equal number of points per level (5 points)**
- B - Twice the number of points in the middle level (10 points)**
- C - Twice the number of points in the top and bottom level (10 points)**

Development of Non-Contact-Type Coordinate Measuring Machine —Measurement of Accuracy of Screw Thread Profile—

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1. Introduction

In order to measure the profile of complex machine elements with high speed and high accuracy, development of non-contact-type coordinate measuring machine is desired. In this study, a measuring machine which can measure coordinates under 4-axes control, is developed and applied to the measurement of profile and dimensions in screw threads.

Factors, which specify the accuracy of screw thread profile, are pitch, major diameter, pitch diameter, flank angle and so on. Among them, some automatic measuring methods have been developed for measurement of pitch accuracy which needs long measuring time^{1, 2)}. Though other factors are important in the inspection of precision lead screw and screw gauge, automatic measuring method have hardly developed. Generally, the measurement of them is carried out by using toolmaker's microscope, three wires method, micrometer and so on. But these methods have some problems such as measurement error due to contact, long measuring time, high skill of operator and so on.

Therefore, in this study, a CNC non-contact-type coordinate measuring machine have been developed on trial, which can evaluate the accuracy of screw thread profile. Some programs are developed for them and the experiments to evaluate

the accuracy of measurement are carried out for M10 screw thread.

2. Photoelectric Probe

Figure 1 shows the principle of the photoelectric probe which is developed in order to detect the profile error of an object with high accuracy. The light from the illuminator is divided into two parts by a half-mirror in order to compensate for the drift of light power, and one part goes to an object to be measured, another part to a PIN photodiode PH2. The real image of the object by the former part of light is focused on the screen (mask) with a pinhole by an object lens. The light which passes through the pinhole is received by a PIN photodiode PH1. This device is called a photoelectric probe hereafter.

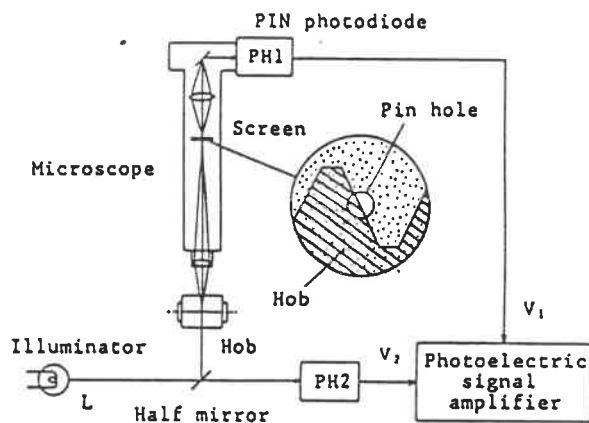


Fig. 1 Principle of photoelectric probe

In the measurement of the profile error, it is utilized that the change of light quantity passing through the pinhole is proportional to the displacement of the image of profile in the pinhole, when the image of profile is in the neighborhood of the pinhole center.

2. Construction of Non-Contact-Type Coordinate Measuring Machine

Figure 2 shows the construction of the whole measuring system,²⁾ which is trially produced by reconstructing a toolmaker's microscope. The second stage (it is called a rotary stage hereafter) on which a head stock and a tail stock are prepared, is mounted on the original X-Y stage of the toolmaker's microscope. The head stock has a stationary center in order to reduce the axial and radial deviations of the measured object during rotation. The measured object, the screw is supported between two centers, and the rotation of the screw axis is transmitted from a gear wheel by a driving plate. The rotational angle of the main spindle, that is the rotational angle of the screw, is

detected by a rotary encoder, the feeding value of the X-Y stage by a laser length measuring system, and the position of the flank in the screw thread by a photoelectric probe, respectively, and they are taken in a personal computer. The main spindle is driven through a spurgear-wheel and a worm gear with high accuracy by a personal computer controlled DC-motor. The X-Y stage is moved by another personal computer controlled DC-motors and screw feeding mechanisms, and the inclination of Z-column (Lens-barrel) is adjusted by an inclining system with a personal computer controlled DC-motor. The profile error is computed from the detected values by the personal computer and recorded in a printer or an X-Y plotter.

3. Measurement Procedure

The origin of X- and Y-coordinates moves by inclining the Z-column. Therefore, a slit fixed on the X-Y stage is used as an absolute origin. In order to avoid the effect of the slit thickness, the origins are defined for the positive and negative X-directions, respectively as shown in Fig. 3.

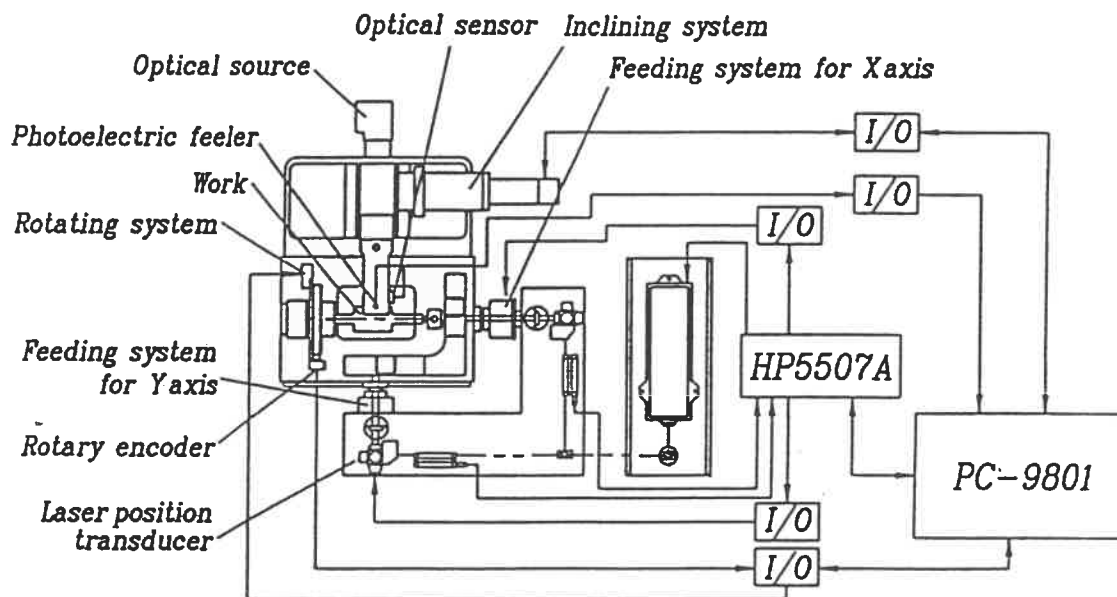


Fig. 2 Construction of the non-contact-type co-ordinate measuring machine

Every time the inclination of Z-column is changed, the origin of the measuring coordinate system is compensated by using the absolute origin of slit. The coordinates of the profile are detected when the pinhole of the photoelectric probe is positioned in the center of the measured profile of screw threads by the computer controlled servo-mechanism.

Figure 4 shows a schematic diagram of measurement procedure for profile and geometrical dimensions of screw threads. Defining the each edge of the slit in the left-top figure as the origins, X- and Y-coordinates are detected for each points considering the lead angle of screw threads. Firstly, the pinhole moves automatically to the crest and its coordinates are detected in turn. After detecting the coordinates of the crest, it moves to a flank and each coordinates are detected by moving in order of arrow, and computes and records the cumulative pitch error, the major diameter, the pitch diameter and the half angle.

4. Results of Measurement

Table 1 shows the specification of screw thread used in this experiment and Figure 5 shows the dimensions. Figure 6 shows the comparison of the result of measurement for the major diameter of the screw threads by this measuring machine with that by SIP universal measuring microscope (UMM). They show a good agreement with each other. The precision of measurement is $0.09 \mu\text{m}$ in standard deviation for repeatability in this measuring machine and $0.40 \mu\text{m}$ in the SIP UMM.

Figure 7 shows the comparison of the result of measurement for the cumulative pitch error by this measuring machine with that by SIP UMM. They show a good agreement with each other. The precision of measurement is $0.17 \mu\text{m}$ in standard deviation for repeatability in this measuring machine and $1.00 \mu\text{m}$ in the SIP UMM.

Figure 8 shows the comparison of the result of measurement for the pitch diameter. The trend of their measured values shows a good agreement with each other. The precision of measurement is $0.36 \mu\text{m}$ in this machine and $1.20 \mu\text{m}$ in the

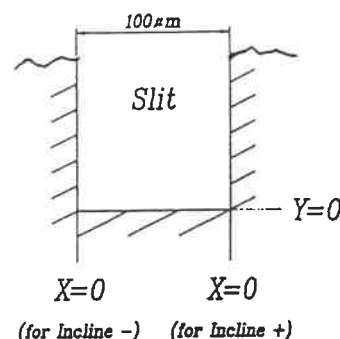


Fig. 3 Compensation of origin

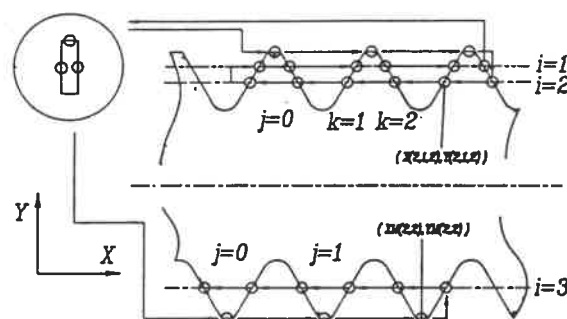


Fig. 4 Procedure of measurement

Table 1 Specification of screw thread

Item	Specification
Nominal designation	M10 x 1.25
Pitch	1.25 mm
Type	Metric fine screw thread
Lead angle	$2^{\circ} 29'$
Half angle	30°

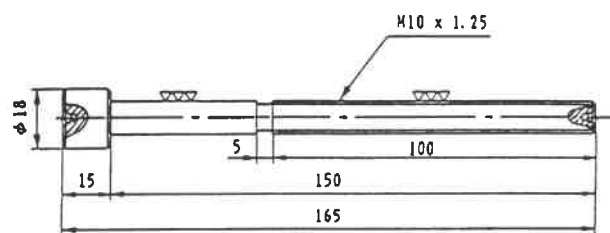


Fig. 5 Dimensions of screw

Table 2 Comparison of measurement accuracy (standard deviation)

Item		This machine	Traditional method	
Major diameter	(μm)	0.13	0.40	(Micro meter)
Half angle	(deg)	0.027	0.17	(SIP)
Cumulative pitch error	(μm)	0.17	1.00	(SIP)
Pitch diameter	(μm)	0.36	1.20	(3 wires)

three wires method.

Table 2 shows the results of measurement for whole factors. In comparison with the results of measurement by the conventional methods, the trend of their measured values shows good agreement with each other and the precision of measurement are improved. In addition, the time used for the whole measurement is about 20 minutes.

5. Conclusion

The main results of this study are summarized as follows;

- 1) A non-contact-type 4-axes controlled coordinate measuring machine is developed.
- 2) The precision of measurement are $0.13 \mu\text{m}$ for major diameter, 0.027 degrees for flank angle, $0.17 \mu\text{m}$ for cumulative pitch error, and $0.36 \mu\text{m}$ for pitch diameter, respectively, in standard deviation.
- 3) The values of measurement in this machine show good agreement with ones in the conventional machines.

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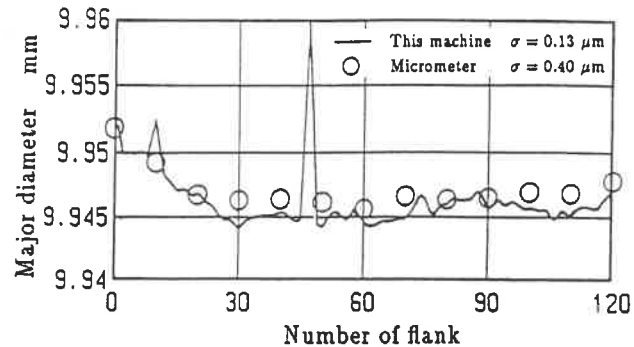


Fig. 6 Major diameter

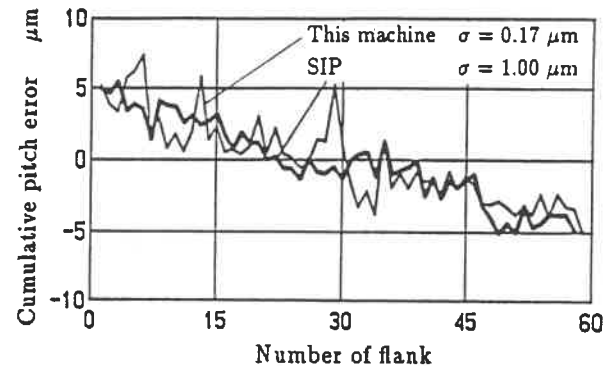


Fig. 7 Cumulative pitch error

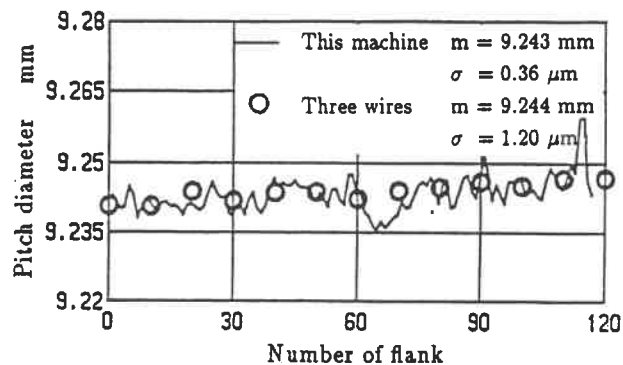


Fig. 8 Pitch diameter

SEM Linewidth Metrology of X-ray Lithography Masks^{1,2}

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Accurate dimensional metrology of the submicrometer gold absorber structures of x-ray masks can be accomplished in the scanning electron microscope (SEM).⁵ This is possible because the x-ray masks present a measurement object that is unique from most other objects viewed in the SEM. This is because the silicon support membrane is, by design, x-ray transparent. This characteristic can be used as a distinct advantage in electron beam-based mask metrology since, depending upon the incident electron beam energies, substrate composition and substrate thickness, the membrane can also be essentially electron transparent. The areas of the mask where the absorber structures are located are essentially x-ray opaque, as well as, electron opaque (Figure 1). Viewing the sample from a perspective below the mask, by placing an electron detector beneath the mask, provides excellent electron signal contrast between the absorber structure and the base membrane (Figure 2). The present technique utilizes a broad acceptance angle detector which is different in concept from other transmission electron (TE) detectors used in the SEM. In this case, the broad angle is used to detect as many of the transmitted electrons as possible (i.e., whether scattered or not) that have an energy above some predetermined threshold which is usually several kiloelectron volts. Then, the electrons are physically filtered both by the signal threshold characteristics of the detector **and** an electron energy filter in front of the detector. Energy filtering of the transmitted electrons excludes the highly scattered and thus lower energy electrons from entrance into the detector. This greatly improves the contrast level over the conventional transmitted electron detection mode for this type of application, and greatly simplifies the required electron beam/sample interaction modeling necessary for edge determination. It is, in fact, this change in electron detection philosophy that makes the present TE approach so attractive for dimensional metrology and inspection of x-ray masks.

Monte Carlo modeling of the transmitted electron signal was used to support this work in order to determine the optimum electron detector position and characteristics, as well as, in determining the position of the edge in the image profile. The Monte Carlo modeling is facilitated in this work, in contrast to the other SEM electron detection modes, because in the transmitted electron detec-

tion mode, the modeling of the electron beam/specimen interaction becomes far less difficult than in the modeling of typical secondary electron images of other opaque objects. Many of the inelastically scattered beam electrons and the low-energy secondary electrons which are complex to model can be excluded from the detector and, therefore, do not need to be included in the calculation. Combining all of these factors, provides a modeled signal that is extremely sensitive to wall slope. As shown in Figure 3, wall slope variation can result in gross differences in the modeled profiles. The comparison between the data from the theoretically-modeled electron beam interaction and actual fitted experimental data is shown in Figure 4 for an effective wall angle of 4° . The theoretical profiles were shown to agree extremely well with experiment, particularly with regard to the wall slope characteristics of the structure obtained from the SEM images and video profiles (Figure 5). Therefore, the theory can be used to identify the location of the edge of the absorber line and thus accurate measurements can be made.

This work provides an approach to improved x-ray mask linewidth metrology and a more precise edge location algorithm for measurement of feature sizes on x-ray masks in commercial instrumentation. The transmitted electron detection mode is also useful in both mask inspection and mask repair, because the high contrast of the image allows for rapid determination of mask defects and high-density contamination particle detection because the transmitted electrons simulate transmitted x-rays. This work represents the first time electron beam modeling has been used to determine the accurate edge location in an SEM image. This also represents an initial step toward the first SEM-based accurate linewidth measurement standard from NIST, as well as, providing a viable metrology for linewidth measurement instruments of x-ray masks for the x-ray lithography community.

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1. Contribution of the National Institute of Standards and Technology (NIST). Not subject to copyright.
2. Certain commercial equipment is identified in this report in order to describe the experimental procedure adequately. Such identification does not imply recommendation or endorsement by NIST, nor does it imply that the equipment identified is necessarily the best available for the purpose.
3. Scientific guest researcher from the Research Institute for Technical Physics of the Hungarian Academy of Sciences.
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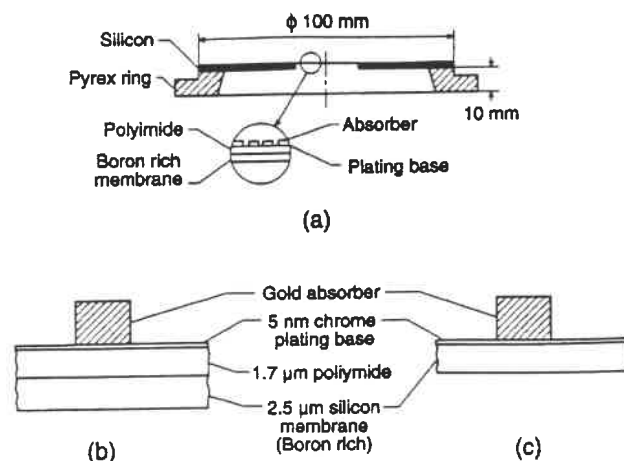


Figure 1. Diagram showing an enlarged cross section of the X-ray mask studied in this work. (a) View of the entire mask assembly. (b) Diagram of the original membrane-absorber cross section. (c) Diagram of the modified version without the polyimide.

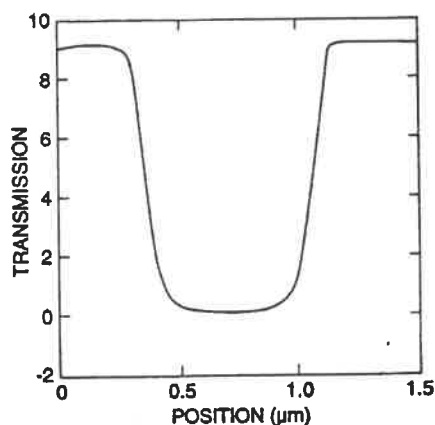


Figure 2. Experimental data obtained from the NIST metrology instrument of a 0.5 micrometer nominal line obtained by using the methods described in the text.

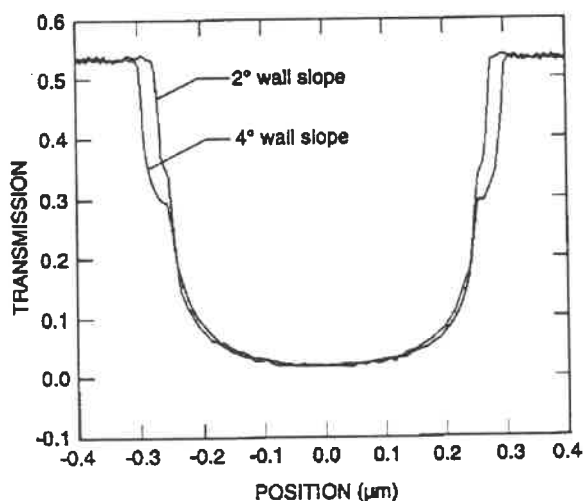


Figure 3. Monte Carlo modeled data for 0.5 micrometer gold line with a 2-degree wall slope and a 4-degree wall slope superimposed.

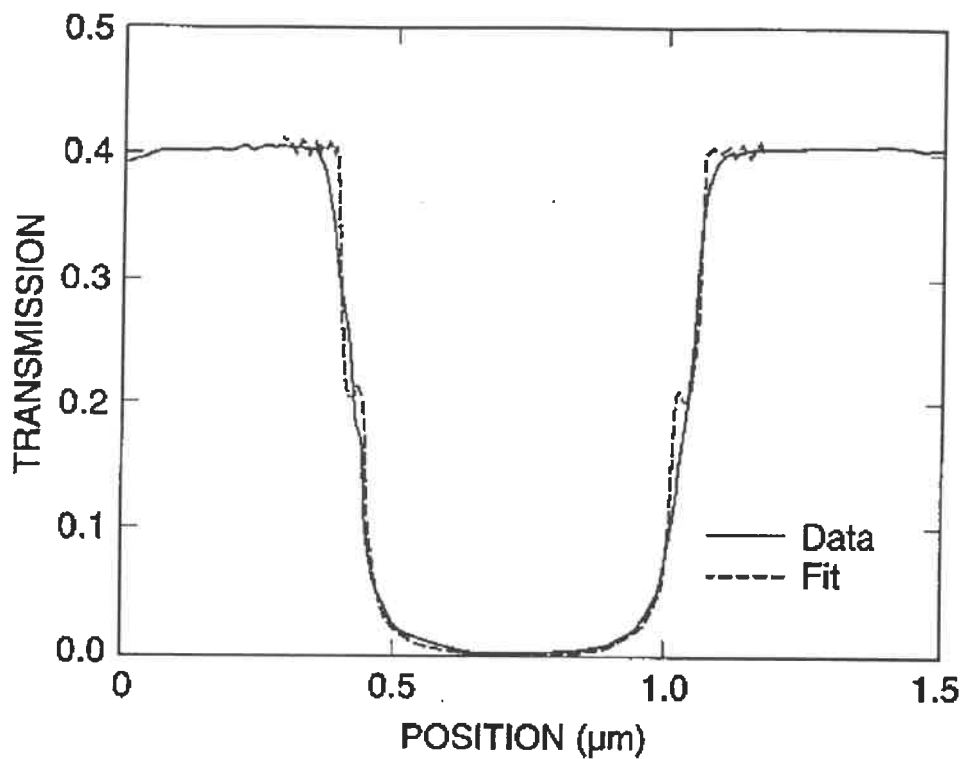


Figure 4. Comparison of the Monte Carlo modeled data for a 4-degree wall slope fitted to the actual experimental data.

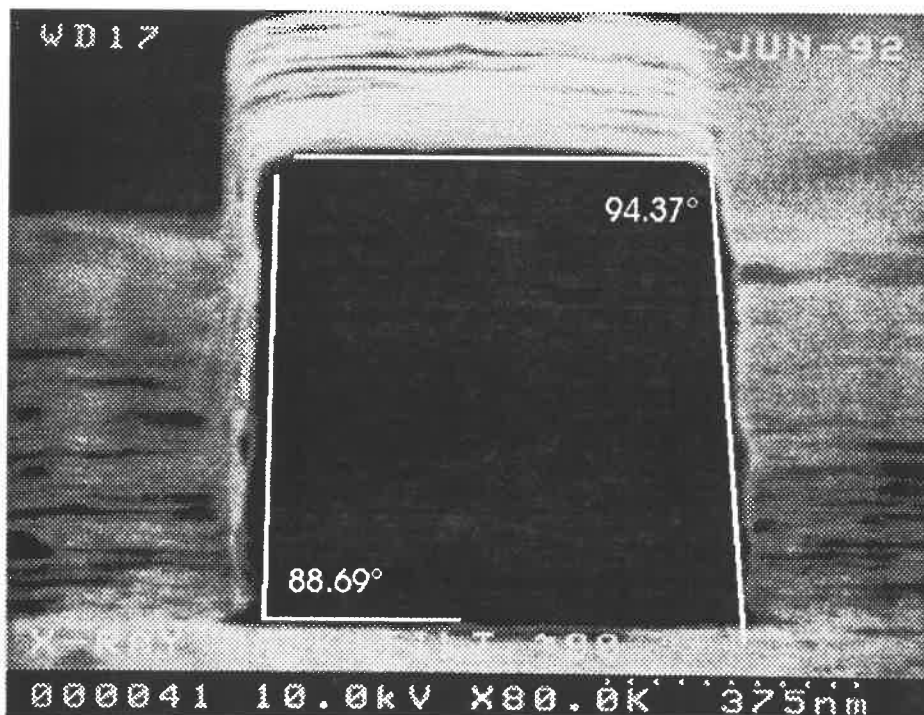


Figure 5. Scanning electron micrograph that has been scanned into an image analysis system demonstrating the approximate 3-degree slope to the sidewalls of the absorber.

Two New Probes for a Coordinate Measuring Machine

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Introduction: We have carried out tests of two new analog CMM probes, both of which are capable of sub-micrometer accuracies. The first is an LVDT-based 3-dimensional contact probe and the second is a non-contact capacitance probe.

These probes were tested on a Moore M48 coordinate measuring machine¹. The three-dimensional accuracy of the machine over small volumes is sufficiently good that it does not significantly limit our ability to test the three-dimensional performance of the probes investigated here. The M48 allows us to perform realistic functional tests of the probe/machine combination.

Probe Testing on the M48: We tested the probes by using them to measure small artifacts on our M48 CMM. The tests were intended to measure both the one-dimensional repeatability of the probes and the full three-dimensional performance as determined by measuring a test sphere or other artifact and inferring probe errors from departures of the measured artifact geometry from the expected (nearly ideal) geometry.

The M48 is a highly precise measuring machine. Errors are small even on an unmapped machine; angular errors are on the order of tenths of arc-seconds and linear errors are on the order of micrometers over the full range of travel of the machine (1.2 meters along the x-axis). Over the small volumes needed to test a probe these errors are not expected to exceed 0.1 micrometer in a cube 10 mm on a side.

Before beginning 3-dimensional probe tests, we did an initial test of the CMM measuring ability for small volumes physically near the point where the artifacts were measured. The test consisted of measuring 49 points in 10 mm x 10 mm regions of planes oriented perpendicular to the three axes and a plane oriented along a body diagonal. Any departure of the measured surface from a plane is indicative of errors in either the CMM or the probe. Random variations were reduced in significance by averaging results from ten runs. Direction-dependent probe errors do not affect the results here because the probe always approaches the plane in the same direction. Results for these four planes indicate that machine errors do not exceed 0.13 micrometer, where the "error" is the full range of deviations of 49 points from a best-fit plane. (The errors ranged from 0.13 micrometer for a plane perpendicular to the y axis down to 0.07 micrometer for a plane perpendicular to z.) Errors of this size are not small enough to be

negligible but should have only modest effects on our results. This test does not detect all possible machine errors; some additional tests to distinguish between machine errors and probe errors are described below.

The LVDT Probe: The LVDT-based probe has a range of 0.1 mm and a resolution of 0.025 micrometer (1 microinch). The probe simultaneously samples the output of three LVDTs to obtain the three-dimensional position of the probe tip relative to its undeflected position. For high-accuracy work we use the probe at deflections far below its full range. At small deflections the probe has a very low contact force, 0.1 gram or less.

The one-dimensional repeatability of the probe/machine combination is approximately 0.04 micrometer (1 standard deviation), as determined by measuring the location of the surface of a gage block several hundred times. This result is comparable to the repeatability of the CMM itself, which is limited by air turbulence along the interferometer scales of the machine. The probe exhibited no mechanical hysteresis; that is, if the probe was deflected along any axis and then made a series of measurements along some other axis or exactly opposite the initial deflection, there was no apparent difference between the first measurement in the series and succeeding measurements.

As a test of the two-dimensional performance of the probe, we measured 49-point circles around the circumference of a 12.7 mm diameter ring gage and around the equator of our 10 mm test sphere. Results for the sphere and ring gage were qualitatively similar, although the sphere results showed a smaller range of values. We attribute the somewhat poorer performance of the ring gage to difficulties we encountered due to dust on the artifacts, a recurrent problem in our laboratory.

Results for a circle measured around the equator of the sphere, averaged over eight runs, are shown in Figure 1. The full range of the deviations of the points from a perfect circle is 0.29 micrometer. (The average range for individual sets of 49 points is only slightly larger, about 0.3 micrometer. Thus random variations in the probe reading added very little to the overall range.) The observed error is significantly larger than estimated CMM errors discussed previously or errors in the geometry of the test sphere.

Thus the results are probably indicative of probe errors, with the exception of a bump 0.16 micrometer in height--perhaps a speck of dust--located about 20° counterclockwise from the y-axis. As a test of this interpretation of the results, we re-measured the equator using a greater probe deflection. The increased

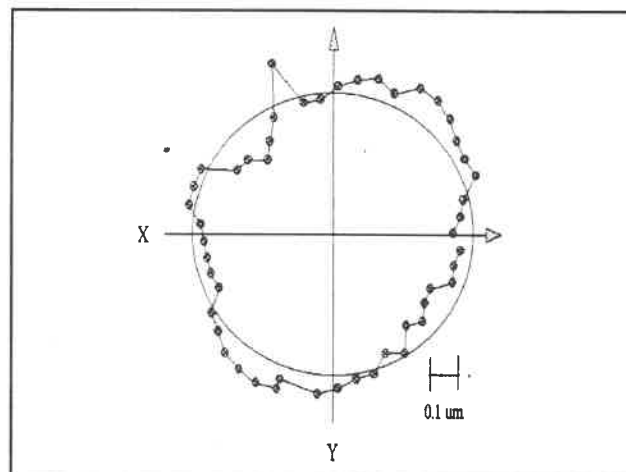


Figure 1: Measurement of a circle on the equator of a sphere.

deflection will not affect errors arising from dirt, CMM geometry errors, or form errors in the test sphere or in the probe stylus, but it will increase the magnitude of other probe errors such as errors in calibration of the probe scales or probe squareness errors. Results of the test show a pattern that is generally a magnified version of the pattern obtained with the smaller deflection; the only exception to this statement is that the magnitude of the 0.16 micrometer bump remained unchanged. The results qualitatively support our contention that other deviations from a perfect circle represent probe errors, but there are still quantitative difficulties with this interpretation that we hope to resolve in the near future.

The three-dimensional point-to-point probing performance of the probe was assessed by measuring sets of 49 points over the upper half hemisphere. A set of ten runs gave an average spread of 0.43 micrometer. The spread of results is reduced 20%, to 0.35 micrometer, if the individual sets of data are averaged; this residual error in the averaged data is a measure of the direction-dependent errors of the probe.

The Capacitance Probe: The non-contacting capacitance probe determines the position of a surface by detecting the capacitance between the surface and the probe stylus, which is a precision ball mounted at the end of a 1 mm diameter hypodermic needle. Figure 2 shows probe output (in arbitrary units) as a function of the distance to a surface, along with a logarithmic fit to the data. The probe sensitivity is proportional to the slope of the curve.

We tested the one-dimensional performance of the probe in two modes of operation. It could simulate the operation of a touch trigger probe, generating a signal to latch the machine scales as the analog reading passes a threshold, or it could be used in true-analog fashion; the analog signal is used to position the probe within less than 1 micrometer of a surface to be measured and the probe's analog output is then read statically. Used in this second mode of operation, the 1-dimensional repeatability of the probe-machine combination is 0.05 micrometer (1 standard deviation). When the probe is used to simulate touch-trigger operation, the 1-dimensional repeatability is about 0.1 micrometer for an appropriate choice of operating conditions. Further tests described below were all carried out in the touch-trigger mode of operation, although we hope to do more tests of analog performance in the future.

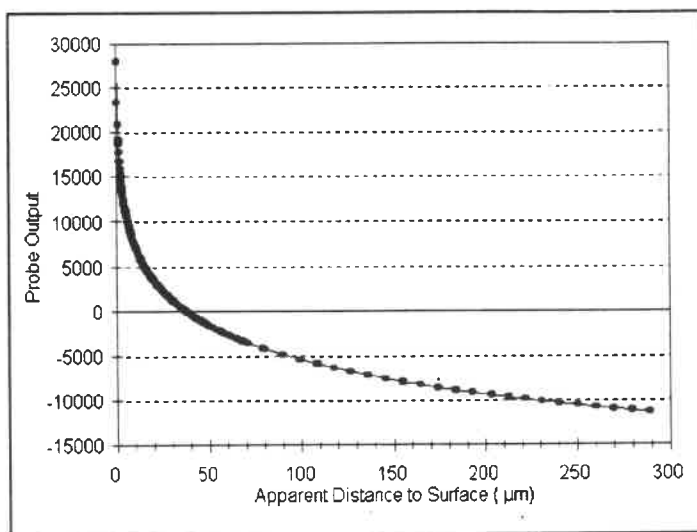


Figure 2: Probe reading vs. distance to a test surface.

The accuracy of this probe depends on several machine-dependent factors. One important consideration on our M48 is the effects of electrical interference generated by our servo motors. Because this

interference is larger than the internal probe noise, it may play a significant role in probe performance. However, it is possible to greatly reduce the effects of electrical interference by triggering the probe very close to the surface of the object being measured. Here the probe sensitivity to variations in the distance to the surface is high. Not surprisingly, we found that probe performance degraded when the probe was triggered at a low threshold, in a region of low sensitivity.

Probe performance also depends on machine dynamics. A particularly surprising result is the dynamic effect illustrated by Figure 3. The graph shows the spread in deviations to a best-fit sphere for nine point measurements of the upper hemisphere, graphed as a function of measurement velocity. The spread in deviations shown here is primarily due to direction dependent errors-- an effective "lobing" error that is a function of probing velocity. Investigations continue to determine the source of this error.

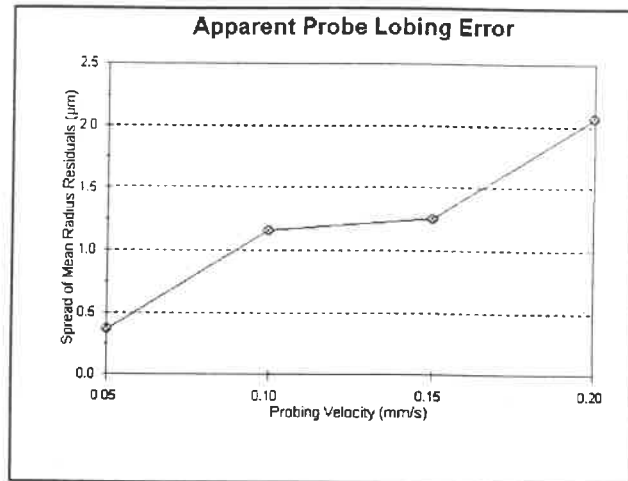


Figure 3: Variation of probe errors with measurement speed.

We have done some 49-point sphere measurements with the capacitance probe in addition to the nine point measurements. At present, we observe a range of the 49 measurements of about 1.3 micrometers at low probing velocities. This error is primarily due to a repeatable directional error; averaging ten runs reduces the range of deviations only slightly, to 1.0 micrometer. It may be possible to improve this value further with a different choice of operating conditions.

Future Work: We are continuing testing of these two probes and expect to have additional results soon. We have begun a study of how the capacitance probe behaves when measuring small features with characteristic dimensions comparable to the probe tip diameter. We have also obtained preliminary results using a commercial touch-trigger probe in order to compare its performance to the performance of the two probes described here.

1. Reference to any product in this paper does not constitute endorsement or recommendation by the authors or their respective organizations nor does it imply that the product is the best available for the purpose.

DESIGN AND TESTING OF A LASER BALL BAR FOR RAPID MACHINE TOOL METROLOGY

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The basic concepts of machine tool error modelling, and instrumentation and methodologies for measurement of parametric errors have been available for over two decades. Numerous studies have shown the value of machine tool metrology for purposes of acceptance, diagnosis, and compensation. Despite these demonstrated benefits, this technology has not gained widespread use in industry. A major reason for this lack of acceptance is the time and expense necessary to perform the required measurements. Characterization of a three axis machining center requires a wide variety of instrumentation, which may cost nearly as much as the machine tool itself, to measure each of the twenty-one error parameters. Highly skilled technicians are needed to perform the measurements, and several days or more of machine down-time may be required to complete the process. The laser ball bar (LBB) was developed to address these difficulties. Laboratory tests have shown that the LBB can measure all of the geometric errors associated with a machine tool (excepting spindle errors) in a nearly automated fashion which requires a small fraction of the time needed for conventional parametric measurement techniques.

The laser ball bar (LBB) consists of a heterodyne laser displacement interferometer whose measurement axis is aligned between the centers of two precision spheres joined by a telescoping tube assembly. It is intended to make precise linear measurements. The spheres ride in trihedral magnetic sockets carried by the tool holder and the machine table. The LBB has characteristics which are similar to the tracking laser interferometer developed by Hocken and Lau and the telescoping magnetic ball bar (TMBB) developed by Bryan for assessment of contouring accuracy over circular or spherical trajectories. In contrast, the LBB features a much longer range of motion than Bryan's TMBB, with relative displacements measured via laser interferometry. This longer range of travel provides the capability of measuring absolute distances over the machine's workspace. The LBB utilizes the telescoping assembly to maintain beam alignment with the moving retroreflector, as opposed to servo controlled mirrors in the tracking laser interferometer. Figure 1 shows two views of the LBB. Light from an interferometric laser source is collected in a polarization preserving fiber optic cable and carried to the polarization beam splitter (PBS). The measurement beam is sent along the telescoping element to the moving retroreflector at the opposite end. This retroreflector returns the beam along the telescoping element where it is recombined in the PBS with the reference beam. The recombined beams pass through a polarizer so that they may interfere. This interference signal is carried from the LBB to a receiver by another fiber optic cable where the displacement information is extracted. A kinematic adjustment system is provided for aligning the measurement beam to the motion of the telescoping tube. The interferometer is used to measure changes in length of the LBB.

The LBB determines tool

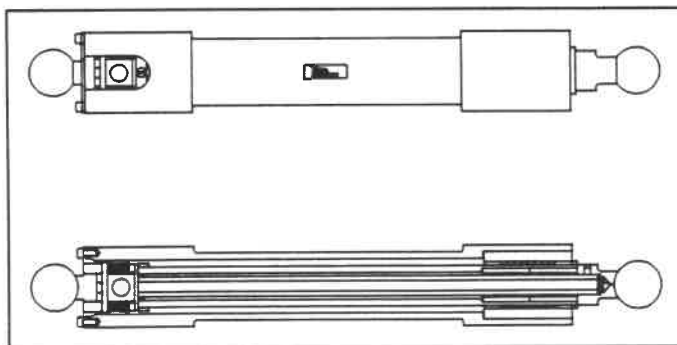


Figure 1. Laser Ball Bar.

coordinates by trilateration, which requires absolute lengths of the sides of a tetrahedron. The displacement interferometer can only measure changes in the length of the LBB. However, if the interferometer readout is initialized to the actual distance between sphere centers at some position of the LBB, then the interferometer readout will continue to give the absolute length of the LBB as it is extended or retracted. The self initialization procedure developed for the LBB utilizes a fixture consisting of three collinear sockets mounted on a thermally stable (slow thermal response) base. The LBB spheres are placed in the middle socket and one of the end sockets. The output of the interferometer is set to zero. The sphere in the middle socket is then moved to the remaining end socket. The interferometer output is recorded. This is the distance between the middle and end sockets. The LBB is then placed between these two sockets and the interferometer output is initialized to this previously recorded distance. If the initialization fixture is rigid and relatively thermally stable, the distance between the two sockets will not change appreciably during the time (10 to 20 seconds) required for initialization.

In use, one trihedral magnetic socket is carried by the tool holder and three similar sockets are placed on the machine table. The initialized LBB is used to measure each of the sides of the tetrahedron formed by these sockets. With the absolute lengths of all six sides of the tetrahedron known, the coordinates of the tool socket with respect to the table can be determined by trilateration. These coordinates are then compared with the machine scale readouts to determine the positioning error of the machine.

The LBB is essentially a precision linear measurement device. When combined with the technique of trilateration, the actual position of the tool can be located relative to a base frame. The difference between the measured position of the tool and the position indicated by the machine's axis scales is the tool point positioning error. For a spatial location problem, such as a 3-Axis CMM, a setup might look like Figure 2. A base plane is defined by the three magnetic sockets on the table. The base lengths of the tetrahedron are measured first with the initialized LBB. Next the remaining sides of the tetrahedron are measured with the LBB and the coordinates computed. If the base sockets are carefully aligned to be parallel with one axis of the machine and the plane formed by the base sockets is parallel with the table surface, then the LBB frame will be parallel to the nominal machine frame and the LBB coordinates can be compared directly with the machine coordinates to determine accuracy.

For semi-automatic metrology, a CNC part program is developed which moves the tool socket to selected points in the workzone. The base lengths are measured first, then the CNC part program is repeated three times, once with the LBB in each of the three base sockets. Suitable data storage and reduction allow the actual coordinates of each of the selected points to be calculated. It is

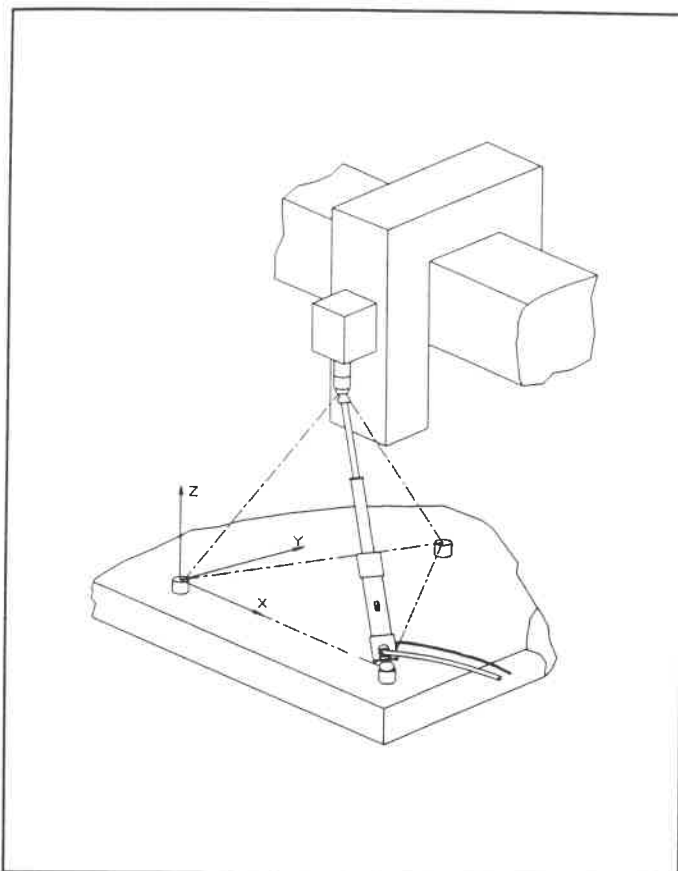


Figure 2. Typical LBB measurement setup.

recognized that this procedure is limited by the repeatability of the machine tool. In practice, however, the repeatability of the machine tool functions as a lower limit for its accuracy. Therefore, this is not considered to be a significant drawback.

The accuracy of the coordinates measured by the LBB is a function of the LBB's length accuracy and the location of the measured point in the workspace. The error associated with the workspace location can be determined by taking the partial derivatives of the coordinate equations with respect to each of the tetrahedron's side lengths. Each of the partial derivatives are multiplied by their respective differential length errors and summed to determine the coordinate error. Assuming a positive and negative error range for each length, the worst case coordinate error can be determined by using the absolute value of the partial derivatives, which insures that the signs will combine for a worst case. A more reasonable expectation of coordinate errors is obtained if the individual length errors are assumed to be randomly distributed and independent of one another. This allows combination of their standard deviations to determine the standard deviation of the coordinate error. For a 2-D workzone analyzed at the worst case location, the maximum coordinate error is 2.5 times the length accuracy of the LBB, and the expected standard deviation is 1.5 times the LBB length accuracy. Therefore, if the absolute length of the LBB is known to within $\pm 1 \mu\text{m}$, the worst case coordinate accuracy is $\pm 2.5 \mu\text{m}$ and the expected coordinate accuracy is $\pm 1.5 \mu\text{m}$.

To be a useful tool for machine tool metrology, the LBB must be capable of measuring tool point coordinates substantially more accurately than the machine tool itself. It was believed that a coordinate accuracy of $\pm 1.0 \mu\text{m}$ would be adequate for metrology of most commercial grade machine tools. Therefore, an initial LBB length accuracy goal of $\pm 0.4 \mu\text{m}$ was chosen. When combined with the trilateration technique, this would insure coordinate accuracy of $\pm 1.0 \mu\text{m}$. To accomplish this, each LBB length error contributor was identified, quantified, and combined in an error budget to guide the design process. The overall system accuracy for the worst case was found to be in the range $+1.43$ to $-2.57 \mu\text{m}$. Using standard deviations, the accuracy is in the range $+0.82$ to $-1.15 \mu\text{m}$. The summarized error budget is shown in Table I.

The repeatability of the LBB was evaluated by repeatedly measuring a fixed distance on the initialization fixture over a 25 minute time period. The LBB was initialized and then used to measure the distance between two sockets on the initialization fixture at both maximum and minimum extension. Repeatability for the minimum range was $0.4 \mu\text{m}$ and for the maximum range was $0.7 \mu\text{m}$.

To assess the overall effectiveness of the LBB for machine tool metrology, the LBB was used to measure the error map of a Mazak QT-28-N CNC Turning center which had previously been parametrically measured and mapped using rigid body modeling. A grid of 63 points were measured over an area of $144 \times 48 \text{ mm}$. The LBB was used to obtain the volumetric errors of the machine five separate times. These five measurement sets required approximately 1.5 hours to complete.

Six parametric measurements are required to obtain the volumetric errors for a lathe if spindle motions are ignored. Two complete sets of parametric measurements were made on the machine. Each set required several days to complete.

The results of the parametric and LBB measurement techniques are compared graphically in Figure 3. The graph shows the mean value of the positioning error as measured by the parametric and LBB methods, superimposed on a grid of their measurement points. The errors have been magnified to make the deformed grids discernible. The maximum deviation between the averages was $3.7 \mu\text{m}$.

ACKNOWLEDGEMENT: This work was supported in part by Grant No. DDM-9017293 from the National Science Foundation.

Table I Error Budget

LBB ERROR BUDGET (all errors in micrometers)		INITIALIZATION ERROR	ERRORS
1	Thermal Growth ...	NO	±0.21
2	Cosine Alignment ...	YES	+0.00 -0.17
3	Sphere—line Misalignment ...	YES	+0.03 -0.00
4	Deadpath and Environmental ...	YES	±0.21
5	Optic's Rotation ...	YES	+0.00 -0.13
6	Sphericity of Ball/Socket ...	NO	± 0.13
7	Initialization Fixture ...	YES	+0.00 -0.003
8	Elastic Elongation ...	NO	+0.00 -0.33
9	Laser Stability ...	YES	±0.016
10	Polarization Alignment ...	YES	±8.2 E-4
11a	Fiber Optic Cable Mech ...	YES	± 0.04
11b	Fiber Optic Cable Thermal ...	NO	± 0.19
SUM TOTAL		+0.30 -0.57	+0.83 -1.43
RSS TOTAL		+0.22 -0.30	+0.38 -0.55
INIT ACC = 2(INIT ERRORS)* + #7		SUM = +0.60 -1.14	
		RSS = +0.44 -0.60	
SYS ACC = INIT ACC + ALL ERRORS*		SUM = +1.43 -2.57	
		RSS = +0.82 -1.15	
* Does not include error #7.			

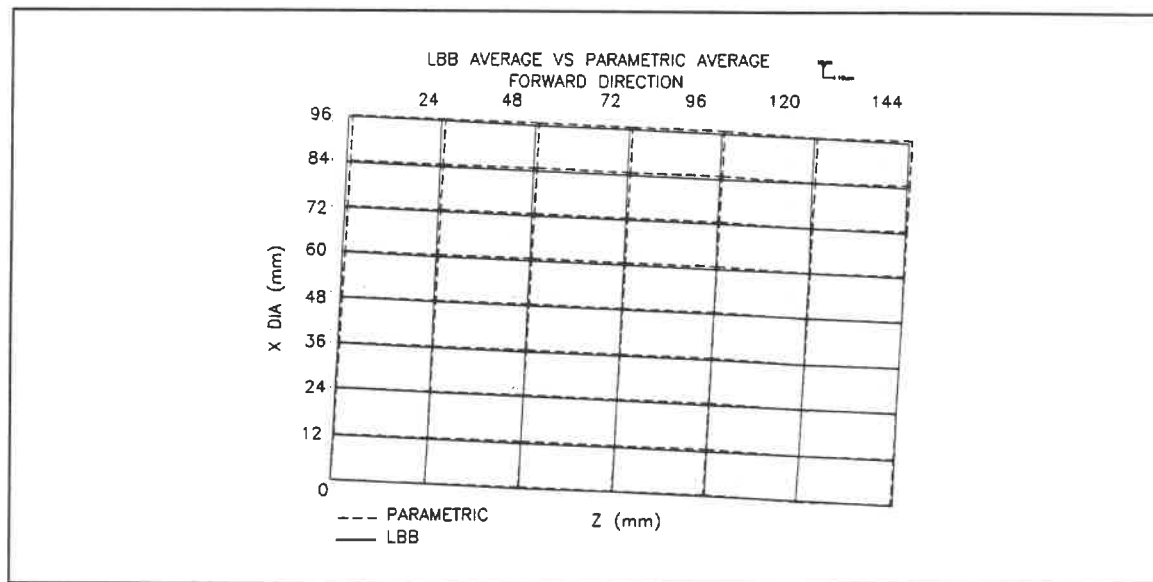


Figure 3 Graphical comparison of parametric and LBB techniques for forward direction.

REPLICATE ELECTROFORMED WOLTER II X-RAY MIRRORS

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Abstract:

Cylindrical mirrors are being electroformed from nickel over an electroless nickel-phosphorous plated aluminum mandrel. The electroless nickel is diamond turned and polished to achieve a surface finish of 10 angstroms rms or better. Gold is then plated on the nickel alloy after an electrochemical passivation step. Next a heavy layer of pure nickel is plated with computer controlled stress at zero to form the actual mirror. This shell is removed from the NiP alloy coated mandrel by cryogenic cooling and contraction of the aluminum to release the mirror. It is required that the gold not adhere well to the NiP but all other plated coatings must exhibit good adherence.

Introduction:

Fabrication of first surface optical components by coating diamond machined aluminum with electroless nickel and subsequently diamond machining and/or polishing the nickel has been reported for several years with typical final component surfaces achieved about 20 - 50 angstroms rms. The nickel - phosphorous alloy preferred is 11% phosphorous (Ni_3P) which provides a dense slightly ductile low stress deposit with sufficient hardness to support an excellent surface finish. This task took this process beyond the conventional processes since the surface requirement was less than 10 angstroms rms and also since the optical surface is on the inside of the component. This required an additional major effort to achieve the actual optical component using the first surface as a mandrel for replication.

The choice was made to produce the optics in-house as much as possible so that the selection of all processes and materials could be controlled. Replication of the surface was at the gold interface to the nickel - phosphorous requiring a debonding oxide film formation without disturbing the 10 angstrom polished surface prior to gold plating. The gold was plated on site from an alkaline non-cyanide electrolyte for three shells and from an off site contracted vacuum evaporation system for the fourth shell. This was followed by a 1 millimeter thick nickel electroformed shell which became the actual optical component. Separation of

all four shells was successful by using liquid nitrogen to cool the composite Al/Ni-P/Au/Ni assembly until the mandrel shrank sufficiently to release from the gold.

Mandrel Material:

The typical aluminum alloy selected for diamond turned optics is 6061 due to good diamond turning response. The material is typically purchased as wrought remelt alloy and contains a large amount of inclusion impurities. No first melt pure aluminum addition 6061 is known to be available in the U.S.

The aluminum mandrel alloy selected for the AXAF-S mirror program was 2024 since it has a higher copper content which promotes less porosity in the plated coating and also since it can be forged. The lower porosity is believed to be due to lower inclusions of iron and also the dissolution displacement of copper on the included iron particles during preparation for plating which in turn permits better coverage of inclusions by the nickel alloy plating process.

Since the parts are very large optics (13 inches diameter by 28 inches long) it was necessary to forge the mandrel blank in at least two steps. The alloy was received in the "F" condition after forging. Since 2024 is not a common forging alloy precautions were taken to assure that no residual stress would be in the material at the time of diamond turning noting that it is not necessary to achieve the highest hardness and strength for this application.

This specification is as follows:

- 1) 21 Hours at 850 deg. F.
- 2) Forge until temperature drops to 700-740 deg. F.
- 3) Reheat at 850 deg. F. for 3 hours.
- 4) Second swage with temperature above 700 deg. F.
- 5) Air cool and ship to MSFC.
- 6) Solution heat treat at 920 deg. F per MIL-H-6088F.
- 7) Boiling water "uphill" quench.
- 8) Precipitation harden 18 hours at 375 deg. F.
- 9) Stress relief after machining 1 hour at 500 deg. F.
- 10) Reverse quench, LN₂ stabilize, steam blast.
- 11) Final machine.
- 12) Post machine stabilize thermal cycle +200 F, -100 F, RT
- 13) Diamond machine allowing for nickel-phosphorous plate.

Electroless Nickel and Gold Plating Process:

The next step is electroless NiP plating. The plating processes were selected on the basis that no cyanide would be used. The NiP plating process used an acid clean, deoxidizer and zincate cycle similar to many other processes

but then a nickel strike from a glycolic acid based solution was used to achieve adherence and coverage of any inclusion sites. The low plating rate was due in part to the addition of sub ppm stabilizer additives which promoted a deposit with outstanding quality performance albeit at the expense of high plating rate. This was evidenced by the complete absence of micro-porosity over the entire 7.5 square foot surface of each of the two mandrels processed. The NiP was heat treated at 320 deg. F for 4 hours. Next the mandrel was sent to the diamond turning and polishing lab where the surface was produced to less than 10 angstroms rms by diamond turning the NiP and polishing with aluminum oxide and either diamond or silica gel.

Next was the passivation and gold plating. NiP deposits were plated using an alkaline sulfite gold process. The passivation process consisted initially of a brief immersion in 2% NaOH followed by 5% K_2CrO_3 without current. A smooth gold deposit was obtained which readily lifted from the electroless Ni in most cases but a very light haze was observed when observed under bright light. Also on occasion during the testing of coupons the deposit was very dark and so poorly adherent that it would peel spontaneously. In order to gain better control of the adhesion a more fundamental understanding of the nature and behavior of oxide films on high phosphorous nickel alloys is under study using potentiostatic methods along with SEM, AUGER, ESCA and AFM.

The formation of oxide films on alloys proceeds differently depending upon many conditions. First, spontaneous oxide formation in air or water occurs for nickel. The time and temperature determine the extent of oxidation with a maximum at about 160 - 170 deg. F. The film formed will generally increase in oxygen toward the surface. The oxide film is generally semiconductive with local sites of higher conductance as a result of film formation defects. Thus the stoichiometry and thickness for a spontaneous film are not well controlled. By reducing the natural oxide cathodically and then forming the desired film anodically better control is achieved. For pure nickel the progression of anodically induced oxide formation is in the order; $Ni(OH)_2$, Ni_3O_4 , Ni_2O_3 and NiO_2 . By adding a complexing agent such as tartrate, the progression of the oxide formation is reduced to two observable steps (i.e., slow) and a more negative and stable reduction of the oxide is observed by cyclic voltammetry.

Nickel-phosphorous alloy samples made anodic in an alkaline nickel tartrate solution have been found to form a uniformly conductive Ni/P/oxide surface. This surface formation can be controlled potentiometrically to form a film less than 10 angstroms thick containing nickel, phosphorous and oxygen which permits a readily deposited and subsequently removable

gold deposit. The presence of carbon was persistent in the outermost film but appeared even in films formed from reagent grade sodium hydroxide indicating that the carbon was adsorbed from the air or from carbonates in the solution. Reduction of the thin oxide film must not occur due to the gold deposition process. The acid cyanide process evolves hydrogen as it plates which reduces the nickel oxide resulting in adhesion. The alkaline sulfite gold process will not readily reduce the oxide film thus permitting deposition with low adherence on the passivated NiP surface permitting separation of the shell later providing that the Faradaic efficiency of the gold plating is near 100% . See Figures 1, and 2.

Nickel Electroforming:

The use of an active sensor to monitor the stress induced in the electroformed nickel shell permitted an on-line method of controlling the stress in real-time. The current density first had to be as uniform as possible. This was mapped using a second monitor and changes in the configuration of the plating equipment were implemented to provide a uniform field and subsequently uniform current density and thickness. The thickness of the part was later measured by CMM to be within 0.001 inch variance over the length of the part. Optical measurements demonstrated that the circularity and optical figure were excellent when proper control was implemented but without this control the quality was not adequate for x-ray applications.

Final Cryogenic Separation of Shell:

After electroforming the nickel shell the part was returned to the diamond turning center and the shell was trimmed to exact length on the mandrel. The part was then cleaned and sent to a cleaner area for separation. The mandrel was hollow and about two inches thick permitting the LN₂ to be pumped into the center thus cooling the part from the inside. Care was taken not to locally heat any part of the shell to avoid thermal difference stress. Thermocouples were used on the outside and when the temperature dropped sufficiently an audible click was heard and the shell was free. This occurred slightly above freezing water temperature at the outside of the shell.

A fixture was built to lift the shell carefully straight up to avoid scratching the surface. After warming the shell was then cleaned and inspected both optically and by x-ray. The focus was better at the longer x-ray wavelengths primarily due to scattering of the x-rays however by the fourth mirror very near flight quality results were achieved.

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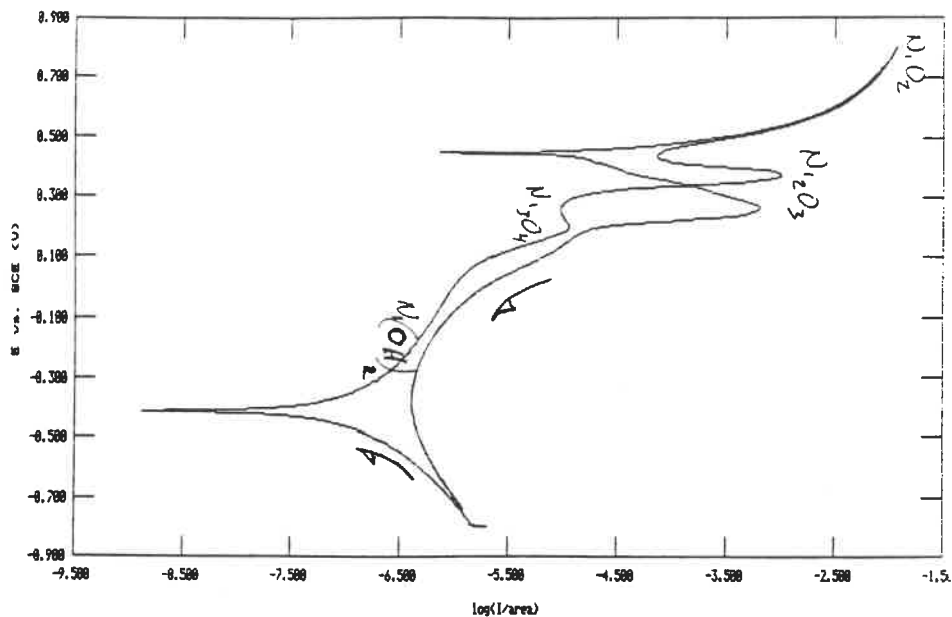


Figure (1) 0.1M NaOH R.T.

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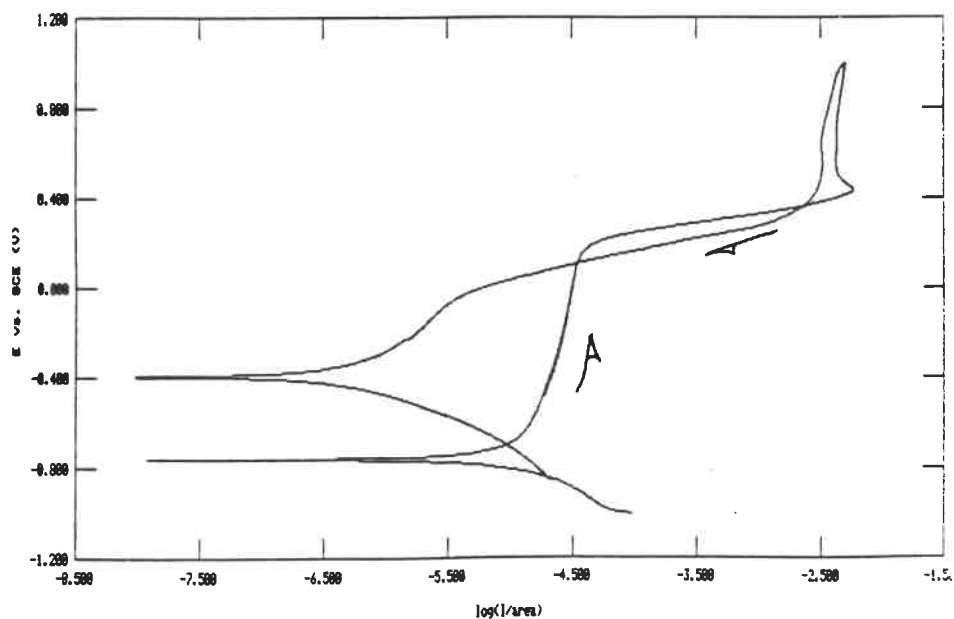


Figure (2) 0.1N NaOH
 0.1M Rochelle Salt
 0.1M Ni⁺⁺

Production of X-Ray Optics by Diamond Turning and Replication Techniques

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Introduction

A demonstration x-ray optic has been produced by diamond turning and replication techniques that could revolutionize the fabrication of advanced mirror assemblies. The prototype optic is a Wolter I x-ray reflector and was developed as part of the Advanced X-Ray Astrophysics Facility - Spectrographic project (AXAF-S). The initial part of the project was aimed at developing and testing the replication technique so that it could potentially be used for the production of the entire mirror array comprised of up to 50 individual mirror shells. The grazing incidence X-ray mirrors for this project are cylindrical shells consisting of parabolic and hyperbolic sections of revolution. The optical surface resides on the inside of the shells which have a wall thickness on the order of one millimeter. This geometry, and the number of mirrors required, mandates the use of rapid and accurate fabrication techniques. Therefore, a replication procedure was chosen that will allow for the fabrication to be done on the outside diameter of the mandrel which is considerably more efficient than working the inside diameter of a thin shell optic. The optic is then formed by a stress free, nickel electroplating process and cryogenically separated. The key components of the fabrication process will be presented with the appropriate testing results.

Replication Process

The fabrication process begins with a large aluminum cylinder that will form the core of the replication mandrel. For this project, two aluminum mandrels were formed to the approximate shape on a tracer lathe and then diamond turned with the optical profiles on the outside diameter. The surface profiles were initially undercut on the radius by approximately 50 to 80 μm to allow for the electroless nickel plating. The basic components of the mandrel are shown in Figure 1. These mandrels were then electroless nickel plated to a thickness of approximately 125 μm and re-turned with the aspheric surfaces. A tongue and groove mounting system was developed to aid in realignment of the mandrel on the DTM. This system worked well and allowed for centering

repeatability to less than 10 μm at the end farthest from the spindle. The turned mandrel is then polished to the required finish and thoroughly cleaned. After polishing the nickel is passivated by growing a thin oxide on the nickel surface and plated with an approximately 1000 $\text{\AA}$ layer of gold. This gold surface ultimately replicates the optical profile and is the reflection surface. Over the gold layer, a special stress free nickel shell is electroplated to approximately 1 mm thick. The mandrel is formed longer than the required optical surfaces to eliminate the edge effects from the polishing phase. Therefore, the optic must be cut to length before separation from the mandrel. The cutting process was performed with a thin diamond blade and a thread grinder attached to the DTM. When the length cuts were complete, the shell was removed from the mandrel with a cryogenic separation procedure. The differential expansion of the shell with respect to the mandrel allowed for a small gap to form between two when the inside of the mandrel is filled with liquid nitrogen. Once removed, the Wolter I x-ray optic is complete and ready for mounting and testing in the stray light vacuum tunnel retrofitted with an x-ray source and detector.

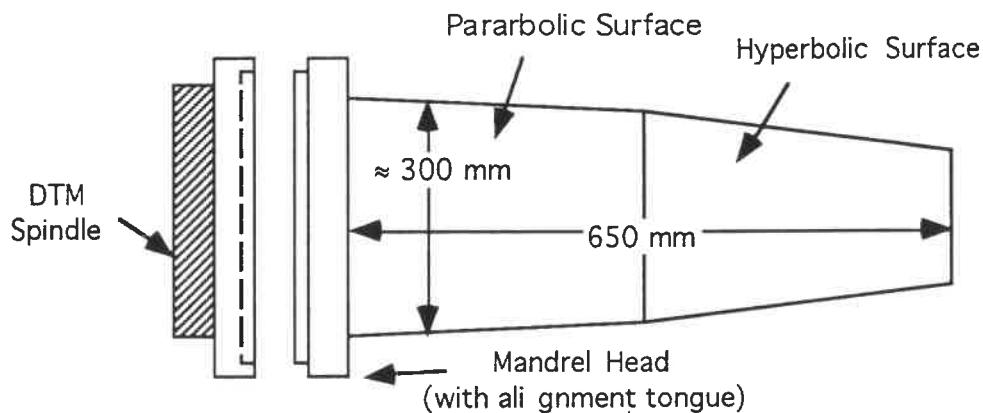


Figure 1 Mandrel for production of Wolter I x-ray reflector.

Diamond Turning

The diamond turning was performed on a Moore M-40 aspheric generator with an oil hydrostatic spindle and roller bearing slides. The first mandrel (FS1) had surface finishes after turning that ranged from $\approx 300 \text{ \AA}$ RMS (Note: all reported surface finish measurements were made with a Wyko 3D at 20X) on the parabolic surface near the machine spindle to $\approx 800 \text{ \AA}$ RMS on the hyperbolic surface at the far end. This variation caused significant problems with the subsequent polishing steps. To reduce the finish to the appropriate levels, the hyperbolic surface had to be worked considerably more and the figure accuracy was degraded with the introduction or exaggeration of some mid-spatial frequency errors (10 to 50 mm in length). Due to the stacked slide configuration of the M-40, the errors in the radial (X) direction inherent in the axial (Z) slide are not corrected with the laser feedback system. This machine

was verified using a Zeiss CMM with a 100 nm resolution. This device proved inadequate due to uncorrected, but systematic, measurement noise and alternative figure measuring devices were considered. After polishing FS2, the mandrel was taken to be measured on a new device at Continental Optics called the Long Trace Profiler (LTP). This device proved quite repeatable and the had a much finer resolution (reportedly around 1 nm RMS over the 1 meter path). Figure 3 shows the five measurements made on the parabolic end of FS2 with the curvature and slope removed. This plot is therefore a map of the mid spatial frequency errors left on the mandrel.

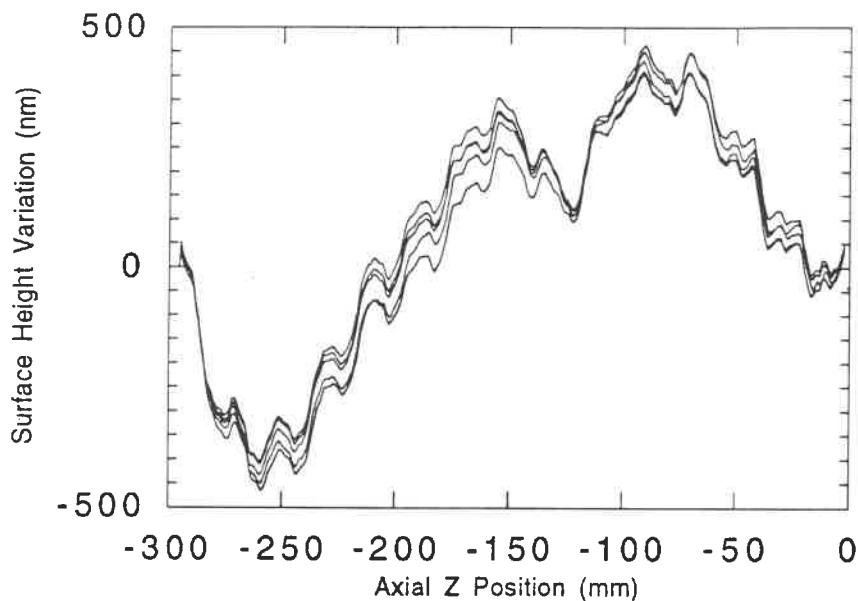


Figure 3 Surface height variation for the parabolic end of FS2 as measured with the LTP.

Conclusion

The diamond turning and polishing operation to form the replication mandrels for the AXAF-S x-ray optics were quite successful. Several problems still exist in the processes and may be correctable for future mandrels. The primary areas of concern are the lack of a suitable thermal environment for the DTM and the inherent machine/part vibration during turning. The thermal environment is probably the main cause of the longer spatial frequency errors and will be corrected when the machine is moved to a new facility. The errors shown in Figure 3 with a wavelength of approximately 20 mm are related to the vibration problems and may be corrected with vibration control measures and potentially a closed loop tool servo system.

was designed to cut normal incidence optics and only motions in the Z direction are referenced back to the machine's metrology frame. The motion of the X slide is referenced to the Z slide and therefore, the X errors in the Z slide remain uncorrected. A map of these errors was subsequently made using a straight edge reversal technique and used to correct the cutting path for the second mandrel (FS2). Figure 2 depicts the uncorrected way errors for the X direction of the Z slide. Additionally, an attempt was made to limit the inherent machine and part vibration for FS2 by altering the rotational speed and using modeling clay as a damping compound inside the mandrel. These changes made a significant improvement in the as cut surface finish on FS2. The Wyko readings were much more consistent over the length of the part and ranged from $\approx 200 \text{ \AA}$ to $350 \text{ \AA}$. This made the polishing operation much easier and resulted in a more accurate figure.

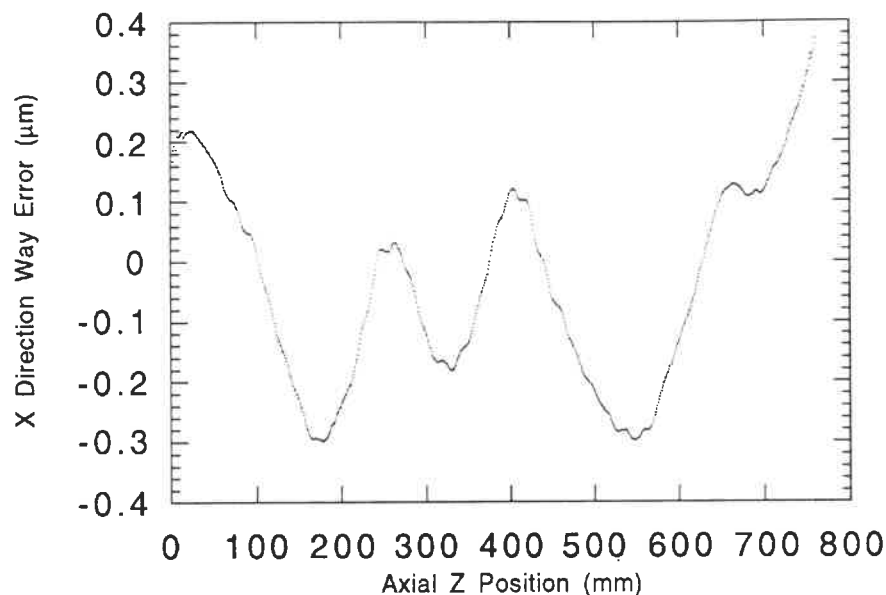


Figure 2 Uncorrected way error in the X direction of the Z slide.

Polishing

The mandrels are polished to the required surface finish on a specially built polishing machine. The polishing compounds were colloidal silica and aluminum oxide. The surface finish of FS1 after polishing ranged from 15 to 20 $\text{\AA}$ RMS. For FS2 the results were much improved and the nominal readings were in the 10 to 15 $\text{\AA}$ RMS range. Most of the time the automated slide of the machine was discarded and the surface was finished by hand. This resulted in a time consuming process that inherently altered the figure. Initially the figure

accuracy depends then on the uniformity of the spacer spring constant of all 1300 spacers. However, the first prototypes built, showed intolerable position errors halfway the stack. Therefore, the stack has been divided into smaller packages of a limited number of blades and spacers. These packages are now compressed and fixed at reference positions every 3 mm by means of precise reference frames. The slit pitch accuracy within one package now strongly depends on the accuracy of the reference frames and, of course, on the uniformity of the spacer spring constant. Next, the blade thickness irregularities are absorbed by an appropriate spacer design. The blades are placed with a small play in spaces created between the adjacent spacers (figure 3). Because thermal expansion is absorbed as well, precision rolled stainless steel could be used for the spacers. As the tungsten blades are very long and very thin, 11 rows of spacers have been applied along the blade length to provide sufficient blade support.

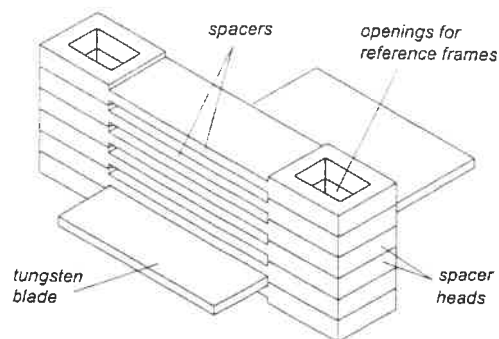


Fig.3. The stack of spacers creates openings for the blades.

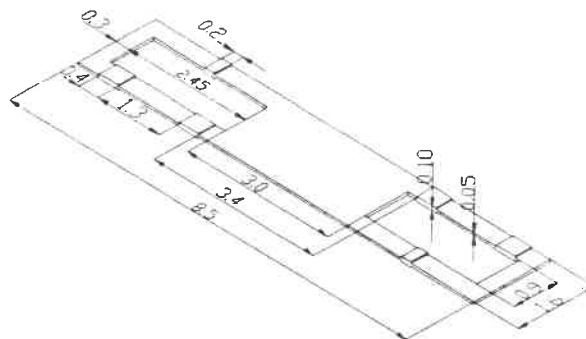


Fig.4. Lay out of the "island" spacer.

Design

A great effort was made to achieve suitable flexibility to the spacers. Eventually, the deflection of special designed spacer heads was used to obtain the desired spring characteristic. The presence of precision etching technique at our laboratory determined strongly the final stepped shape of the spacers (figure 4).

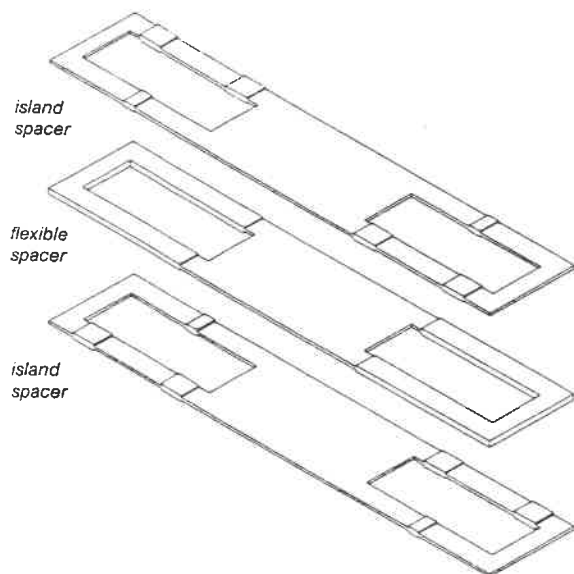


Fig.5. Configuration of the "flexible" and the "island" spacers.

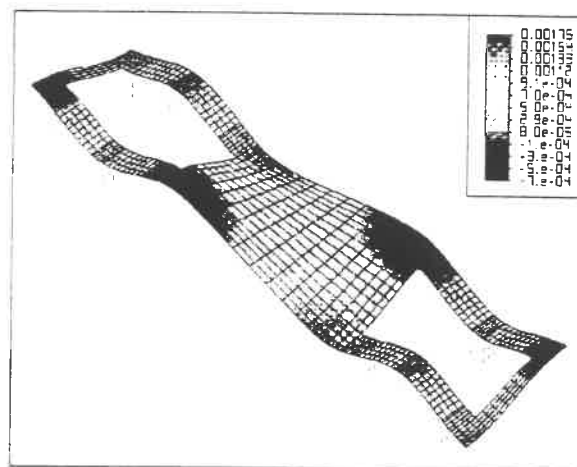


Fig.6. Outcome of the finite element deflection of the optimised spacer.

DESIGN AND MANUFACTURE OF SLIT PATTERNS FOR GAMMA RAY IMAGING

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Introduction

Many astronomical objects emit gamma rays. For some processes, like solar flares, gamma rays carry the most direct evidence available for the roles of accelerated particles. Unfortunately, gamma rays cannot be imaged in a classical way because no lenses exist for this wavelength range. However, it is possible to measure Fourier components of the images by observing gamma ray sources through a limited number of fine patterns of slits placed at a mutual distance of several meters (see figure 1). For reasons of the necessary gamma ray absorption by the material surrounding the slits the patterns have to be realised in relative thick (3 mm) high-Z material like gold, tungsten or tantalum. In order to achieve the desired high spatial resolution of the images, the slit geometry must meet very high precision goals. The finest pattern must have slits of $50\text{ }\mu\text{m}$ wide at a pitch of $100\text{ }\mu\text{m}$. The absolute position accuracy of the slits must be in the order of a few micrometers within an area of $130\text{ mm} \times 130\text{ mm}$. The position requirements must be maintained in the hostile environment of temperature variations and mechanical vibrations during satellite launch.

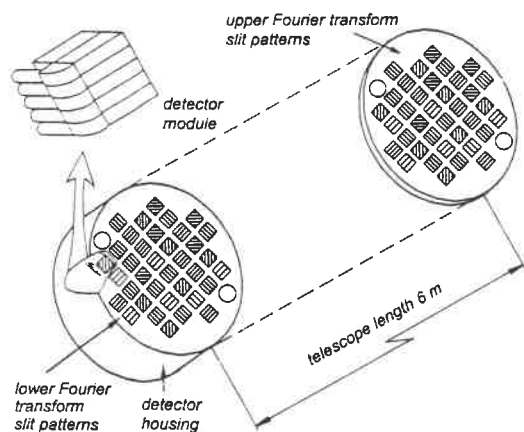


Fig. 1. Configuration of slit patterns within the gamma ray telescope.

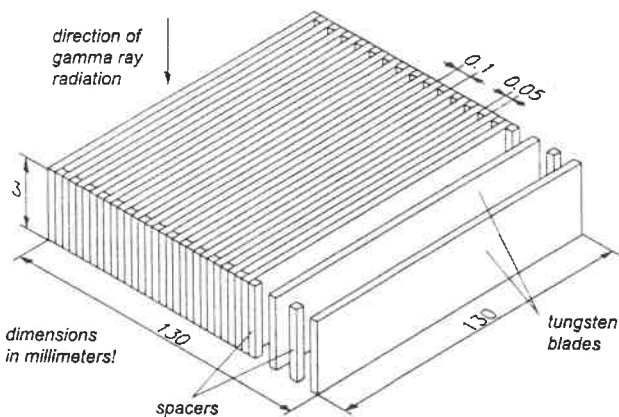


Fig. 2. Slit patterns composed of blades and spacers.

Approach

Under NASA research contract, these slit patterns have been developed. It was obvious that these slit patterns can not be made by known machining technologies. Even with the new x-ray deep lithography (LIGA technique) it was not possible to create the demanded slit geometry given the thickness of 3 mm. The decision was made to realise the slit patterns by stacking 1300 long thin tungsten blades with small spacers in between (figure 2). However, the high accuracy of slit width and slit pitch could not be obtained by ordinary stacking of the parts due to the limited thickness accuracy of the rolled materials. Even a very small thickness deviation of tenths of a micrometer from the nominal value could result in a huge position error.

A promising solution turned out to be the application of spacers with the character of a spring in the stacking direction. This solution can avoid the cumulative stacking error. If each spacer can be compressed by a few micrometers, the average slit pitch can be adjust to the desired value very accurately simply by compressing the whole stack of spacers and blades. The absolute slit pitch

The spacer configuration consists of alternate stacking of "flexible" spacers and "island" spacers (figure 5). The presence of islands causes, after compression of the spacer stack, a deflection of flexible spacers.

The final spacer design was optimised using ALGOR™ finite element analysis. The final spacer configuration with the cross symmetrical deflected shape (figure 6) gives the best possible blade position.

The proper position of the 3 mm spacer packages within the whole structure of the slit pattern is dictated by the reference frames. There are 11 parallel reference frames with a mutual pitch of 13 mm. Within these frames reference planes are machined with a very high accuracy (figure 7). Electric discharge machining has been applied for their manufacture. Actually, the position of the compressed spacer package is determined by two wedges placed on the opposite sides of the 3 mm package. These wedges are pushed against the reference planes by the spring force of the compressed package. To hold the wedges in place and to obtain a seamless stringing of adjacent spacer packages, it was necessary to alternate the spring force (and therefore also the spring stiffness) of the neighbouring packages. Therefore, there are two different sorts of packages, stiff and less stiff. The stiffer packages hold the wedges against the reference planes, while the less stiff packages in between do not affect the proper wedge position. The difference in the stiffness was achieved by adjusting the size of the islands. Stiffer spacer stacks consist of spacers with slightly larger islands. Figure 8 shows the alternate packaging configuration.

Thermal variations could destroy the pitch accuracy, therefore the reference frames have been made from "invar" - a zero thermal expansion material. All frame parts parallel to the blades possess the same expansion coefficient as the tungsten blades. Special attention also is paid to the mutual alignment of the 11 parallel reference frames as the blades must remain very straight. Integrated spring based micro manipulators allow an adjusting accuracy of $\pm 1 \mu\text{m}$.

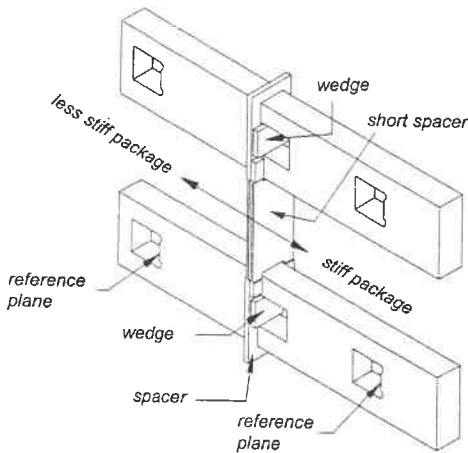


Fig.7. Reference frame with the accurately machined reference planes (stops).

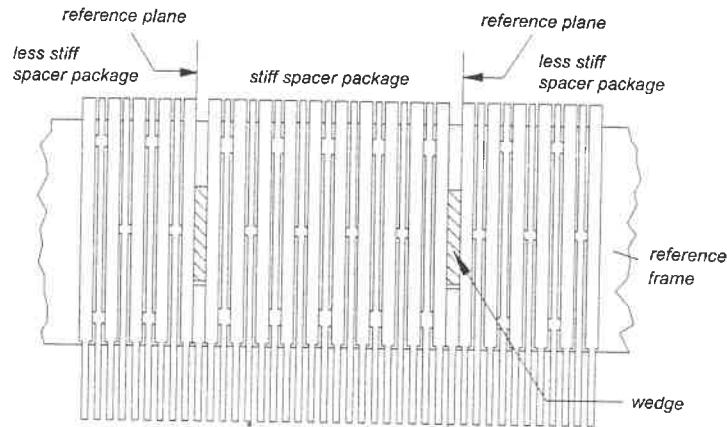


Fig.8. Stiff and "less stiff" spacer packages alternate within the spacer stack.

Manufacturing

The spacers (total number: 14300) were made by a precision metal etching technique. All steps in this technique have been optimised in order to achieve high dimensional accuracy. As can be seen from figure 4 there are some critical dimensions to be kept under control. The most critical is the thickness of the spacer middle region. This thickness was calculated to be $44 \mu\text{m}$ with a allowable tolerance of $\pm 2 \mu\text{m}$. The step shaped spacer thickness could only be achieved by a multistep etching technique. Positive photo resist has been used as a selective protective layer during the etching process. The symmetrical double sided etching starts at the spacer outline (figure 9). After a certain calculated depth has been

reached, new areas of photoresist are exposed to determine the areas with the reduced thickness. The etching continues till the desired thickness is reached. To meet the tolerances, very high uniformity of the etching process must be maintained by controlling many parameters of the etching process. Extended experiments have been performed to find the most suitable machining technique for the heavy metal blades. Finally, the blades were ordered to very tight tolerances at a New Jersey based rolling company. The blades have been rolled from a round wire, therefore no sharp edges or blurs can disturb the stacking accuracy. Assembling of the full size pattern has been performed in a clean room environment. Figure 10 shows the device enclosed in a transparent transportation case ready for shipping.

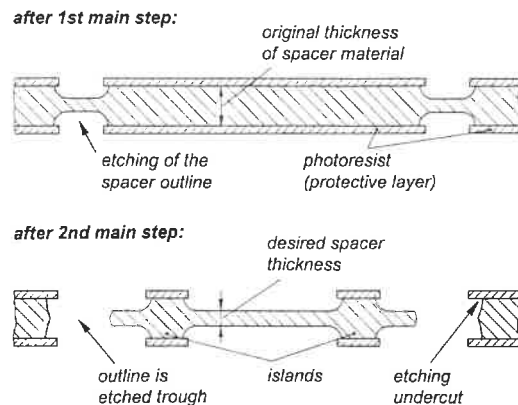


Fig.9. The multistep etching technique.

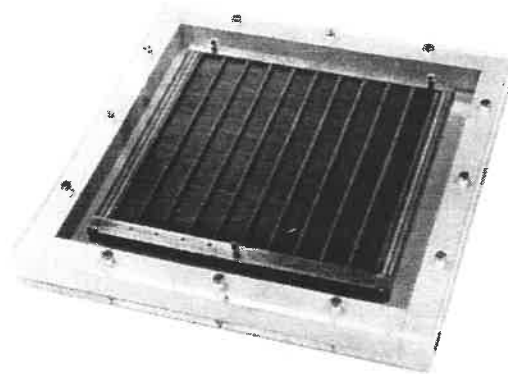


Fig. 10. The slit pattern ready for shipment.

Results

Extended measurements of the assembled slit pattern has been performed using optical measuring techniques. More than 5000 measurement points were taken in account to determine the final accuracy. The standard deviation of the most important slit parameter, the absolute slit position, is less than $8\mu\text{m}$. Figure 11 shows the results for the first 400 slits. A more detailed profile of the slit position error can be seen in figure 12. As expected, in the neighbourhood of the reference planes the position error returns periodically to a very low value.

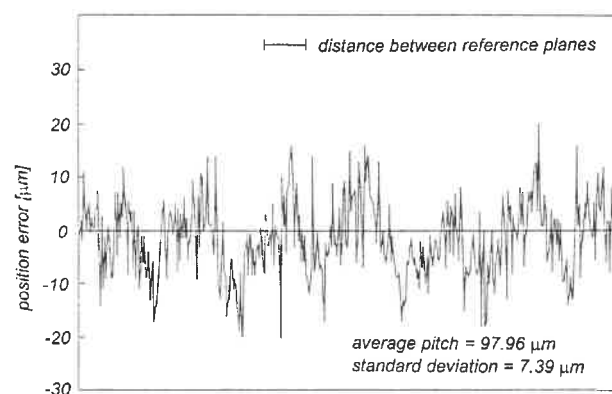


Fig. 11. The slit position error of the first 400 slits.

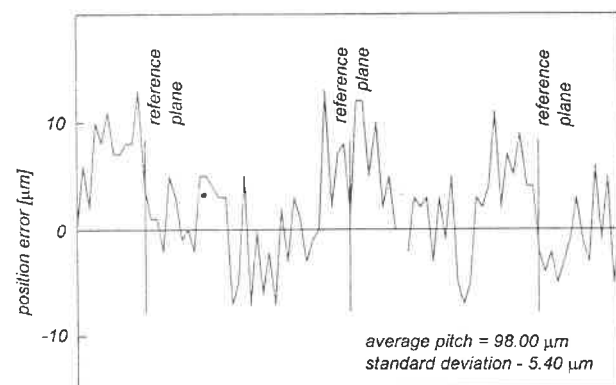


Fig. 12. Detailed graph of position errors at reference planes (every 3 mm).

The very promising results of this slit pattern allow us to believe that the same principle of assembling technique can be used for patterns with a considerably smaller slit width. First steps were already taken to develop slit patterns with $21\mu\text{m}$ slit width.

PROSPECTS FOR X-RAY OPTICS FROM ATOMIC LAYER GROWTH

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A new type of precision engineering shows promise for fabricating x-ray mirrors and diffractors. Atomic layer growth (ALG) has construction elements (layers) that are only one atom thick. Thinner layers are impossible, so we have reached ultimate precision in this dimension. However, atomic layer growth is far from a turnkey operation at present.

Stimulus for the development of ALG came partly from the unusual requirements that x-ray spectroscopy places on optical engineering. Only diffractive and reflective optical elements can be used because x rays do not bend as they pass through prisms or lenses. Optical engineers typically specify the rms roughness of mirrors to be less than 1/20 of the wavelength of the incident radiation. For soft x-ray wavelengths (0.1-10 nm), this implies atomically smooth surfaces. However, even if a surface is smooth, it reflects only a very small fraction of incident x-rays. Reflectance can be increased to some extent by depositing multiple layers in a stack, alternating a dense layer with a less dense layer. If layer-pair thickness (d) matches the wavelength (λ) and grazing angle (θ) of incidence, constructive interference occurs and enhances reflectance. This relationship is expressed by the Bragg law:

$$n\lambda = 2d \sin\theta.$$

The positive integer, n , is the order of diffraction. Reflectance generally decreases dramatically as n increases. Other more complex equations relate the percent reflectance to the following factors:

1. Interface roughness which degrades reflectance
2. Atomic diffusion at interfaces which degrades reflectance
3. Large number of layers which improve performance up to a point (20 to 1000 pairs, depending on wavelength)
4. Optimum optical properties of the materials in the layers which can improve performance for most applications

These design features challenge the optical engineer to solve the problem by using new materials to fabricate many layers of constant thickness with atomically smooth and abrupt interfaces.

Physical vapor deposition methods (sputtering, evaporation, etc.) and chemical vapor deposition methods make useful multilayers, but they are inherently prone to roughness and diffusion because of their stochastic nature. They place atoms in a somewhat statistical manner in both space and time. As an alternative, we have used atomic layer growth to deposit a sequence of layers, each of which is one (and only one) atom thick. This allows precise, digital control of layer thickness, and ALG usually repeats the smoothness of the substrate. The layers may reproduce the crystal structure of the substrate, in which case the process is termed atomic layer epitaxy (ALE), a method which is often applied in the

semiconductor industry for growing strained layer superlattices and other electronic structures [1,2].

The atomic layer processes have much in common with chemical vapor deposition [3]. The substrate is maintained in a hot, relatively broad temperature range (e.g. 390-460° C) [4]. A pulse of gas (e.g. zinc vapor) is allowed to enter the deposition chamber, and it comes to equilibrium in 1 to 30 seconds by depositing a monolayer of physically absorbed molecules on the surface of the substrate. Both pressure (P) and temperature (T) must be matched within limits for this to occur. A new material may have a different matching P/T range, but it will probably overlap the range of some other materials. The monolayer may react (form covalent chemical bonds) spontaneously with the substrate, or it may require the entry of another gas to produce chemical bonds.

During the next phase of the cycle, the remaining vapor is removed from the chamber by evacuation or displacement with an inert gas. However, the monolayer remains. Then a new type of reactive gas (e.g. sulfur) enters at the proper pressure for depositing a monolayer. In this particular case, reaction occurs so that a monolayer of the chemical compound ZnS is formed. The unreacted gas is removed and the entire cycle can then be repeated as many times as desired. The same type of ZnS layer can be obtained with a number of different reactant gases for the Zn and the S (e.g. ZnCl₂ and H₂S). Proper gas selection simplifies some processes and permits deposition of refractory materials (e.g. deposition of solid tungsten from tungsten hexafluoride gas).

There are a number of options that are attractive in principle, but difficult in practice.

1. Deposit one or more layers of one compound (e.g. ZnS) and then deposit one or more layers of another (e.g. CdSe).
2. Deposit one or more layers of one element (e.g. W) and then one or more layers another (e.g. B).
3. Use light to photochemically deposit a monolayer only in the illuminated areas.
4. Combine any of the previous steps to produce a sequence of layers of arbitrary digital complexity.

Most of the difficulties come under the headings of contamination, safety, and compatibility. Contamination by oxygen or water disrupts most ALG processes. In addition, substrates must be scrupulously clean. Most of the gases are toxic. Each must be pumped out on every cycle, and then safely stored or reacted for disposal. Workers must not be made vulnerable to leaks in the system or to failure of parts (e.g. frequently used automatic valves or cracks in quartz reactor vessels).

Compatibility covers a multitude of factors, especially for atomic layer epitaxy. The growth should be started on a single-crystal substrate (e.g. silicon wafer) that has the proper crystal lattice form and size. If an amorphous material (e.g. smooth glass) is used, crystals will nucleate at various locations and grow together to form a mosaic pattern. Spontaneous surface reconstruction (which minimizes dangling bonds) should be avoided if possible. At points where there is a change in the stack from one crystalline material to another, there must be a good match between the crystal structures and the dimensions of the unit cells. However, it is sometimes possible to orient one unit cell in such a way that it fits a different underlying crystal structure. The materials must be chemically compatible and available in forms (gas or volatile condensed phases) that conform to process requirements. In addition,

they should have similar coefficients of thermal expansion to prevent excessive intrinsic stress upon cooling. Although their deposition pressures can be very different from one another, the deposition temperature ranges must overlap or the temperature must be adjusted within acceptable limits during each cycle. Multilayer deposition is impossible if the deposition temperature of one material is so high that it removes a formerly deposited layer.

There are other atomic engineering processes, but ALG has advantages over them. Molecular beam epitaxy, which is used for research in the semiconductor industry, is limited to fewer materials as reactants; it requires expensive, ultra-high vacuum equipment; it is a line-of-sight method (won't work inside pipes, etc.); and it is very slow. ALG is much faster, but still slow by conventional standards such as electroplating. The maximum rate for ALG is about one monolayer (0.1 nm) per second. This implies a reaction time of at least 2.7 hours per μm of thickness, or 70 hours/mil. A compensating factor is the ability of the reactant gases to penetrate beyond line-of-sight paths. A reactor could contain stacks of parts separated by thin gas paths so that many square meters of area could be coated in a reactor of moderate volume. Coating would occur in vias and even tortuous holes.

Atomic engineering has also been accomplished by pushing atoms around on a surface with the tip of a scanning probe microscope.

A popular showpiece micrograph reveals the letters IBM written with about 40 atoms. It probably took hours to push them into position. A square inch of a monolayer of iron contains about 4.6×10^{16} atoms. Arrangement of this layer at 40 atoms per hour would take a very long time. It is not clear how this process could lead to manufactured products.

Using a reactor that we have constructed, we have deposited several different elements (W and B) and pairs of elements that form chemical compounds (ZnSe and SiO_2). Our W/B structure showed some evidence of the desired x-ray reflection, but we must modify our methods to achieve the full capability of ALE to produce useful x-ray optical elements.

Very few pairs of materials have crystals structures that match adequately for heteroepitaxy by ALE. Mismatch creates stresses that result in elastic strain, crystal defects, and even peeling from the substrate. However, the number of compatible pairs can be greatly enlarged by the use of dilute alloys to help adjust lattice parameters and by limiting the thickness of each material. Stress and chemical composition affect layer thickness in ways that can be estimated from the Poisson's ratios and from Vegard's Rule[5].

The edges of multilayers provide interesting mechanical, optical, and even chemical potential. If the edge of a multilayer stack is ground at an angle to the planes of the layers, and alternate layers are chemically etched, a series of cantilevers results. Each has a smooth top and bottom, so the group can be used as an optical etalon. Entirely etching away alternate layers will leave a series of ribs which can be held in place by a frame. The minimum gap between ribs could be as small as an atomic diameter (about 0.1 nm), but Van der Waals forces would provide strong attraction from rib to rib and could cause collapse of any structure not having a condensed phase between the ribs. The minimum rib width appears to be larger (about 2 nm) because surface tension pulls thin structures into globular forms. Atom mobility governs the rate of this process, and it can be slow at temperatures of less than $2/3$ the absolute (Kelvin) melting temperature of the material, provided no bombardment (e.g. ion milling etc.) occurs. Multilayers formed around a mandril or fine wire could be used (possibly after etching) for optical zone plates. The

edges of multilayers have also be found to show excellent ability to catalyze certain chemical reactions [6].

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Precision Ion Figuring System Development

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Introduction

An important step in the fabrication of an optical component involves the expensive and time consuming process of imparting a precise contour on the optic. Ion beam figuring modifies this contour by removing material through the impingement of a broad beam of accelerated neutral particles. This technique provides a highly deterministic method for the final precision figuring (or correcting) of optical components with advantages over conventional methods. The repeatability of the process allows the possibility of single step figuring, resulting in significant time and cost savings. Unlike grinding, polishing and lapping, ion figuring is non-contacting and so avoids several problems including: edge effects, tool wear, and loading of the work piece. It has previously been demonstrated that ion figuring is effective for the correcting of large optical components [1][2].

The work discussed here is directed toward the development of the Precision Ion Machining System (PIMS) at Marshall Space Flight Center. This system was designed for the processing of small (≤ 10 cm diameter) optics. The development of the figuring system has proceeded in three steps: 1) the development of control algorithms for determining parameters to guide the ion beam; 2) the use of a simulation to verify the methods and to identify important issues and parameters that are critical for the proper operation of the system 3) the integration of the controlling algorithms with the other components. Experimental testing of the system will then follow and includes the figuring of chemically vapor deposited silicon carbide (CVD SiC) and fused silica samples. An emphasis will be placed on CVD SiC that have been previously fabricated by ductile-regime grinding.

Earlier research at this facility has indicated that the removal of subsurface damage in the preceding step is critical to maintaining surface quality [3]. The experiments involved ion machining of CVD SiC samples that were polished or ductile ground to specular or near-specular surface finish. The polished samples exhibited significant degradation of the surface roughness while the ductile ground samples did not. It is believed that this was due to subsurface damage left on the polished samples.

Control Methods

The ion figuring process involves bombarding a surface with a temporally and spatially invariant beam of neutralized particles, typically argon. Material on the surface is consequently sputtered away. The rate and distribution (profile) of material removal by the ion beam is kept constant and specific figures or corrections are achieved by rastering the fixed-current beam across the workpiece at varying velocities. The figuring process relies on obtaining an accurate figure error map of the workpiece surface. The solution process consists of choosing a pattern the beam traverses and determining the rastering velocities along that path. The removed material can be computed by convolving the ion beam's fixed removal profile and the rastering velocity as a function of position. The determination of the appropriate rastering velocities from the desired removal map and the known ion beam removal profile is therefore a *deconvolution* process.

A unique method of performing this deconvolution was developed for the project based on research on singular expansions of filters [4]. The solution to a convolution equation can be expressed in terms of derivatives of the known function and inverse moments of the filter. The deconvolution algorithm requires information on the moments of the ion beam removal profile and a series expansion of the surface map of the work piece. A unique and stable solution is then directly available. The assumption in this algorithm is that the expansion represent the desired removal map. There is no special consideration to the geometry of the workpiece in the x-y plane. The solution provided is exact so the accuracy of expansion fit to the desired removal is the measure of the solution's accuracy.

Simulation Results

A simulation of the process was developed to verify the algorithms and to identify important issues and parameters that are critical for the proper operation of the system. All simulations involve data from an actual 8 cm flat sample measured with a ZYGO interferometer. The process of figuring the sample to ideal flat and spherical contours is then attempted by performing a discrete convolution of the ion beam removal profile and the computed solution. The resulting reduction in the RMS difference between the actual surface contour and the desired surface contour is used as the measure of the efficiency of the process. The sample has an initial error of 234 nm RMS from a flat plane, and 113 nm RMS from a 400 m radius sphere.

The expansions representing the error contours (removal map) are polynomial series. The higher the order of the series, the more accurate the solution. Initial experiments were performed to evaluate the effect of the order on the performance of the system. The number of terms used in the series expansion representing the removal map was increased and corresponding increases in the efficiency of the process were observed. The results are shown in Figure 1 as

the final RMS difference from an idealized flat plane.

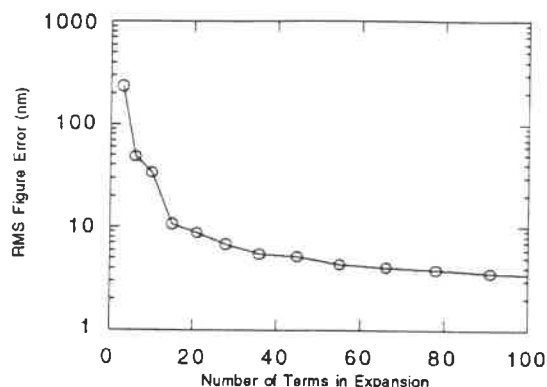


Figure 1 Results of simulation to determine effects of expansion order.

In Figure 1, the first point represents the initial RMS difference at the three expansion terms required to define a plane. The final RMS difference is directly related to the accuracy of the expansion fit. Using fifteen terms the error was reduced to 10 nm RMS and after one hundred terms to 3 nm RMS. Above approximately fifteen terms there is no significant increase in figure accuracy and 10 nm RMS is well below the accuracy of the original data. Other simulations were performed to determine the effects of systematic and random errors. The errors were introduced through the variations in the model of the beam function. The ideal beam function is modeled as a Gaussian distribution defined by two parameters: the width (λ) and the amplitude (Γ).

$$H(x,y) = \Gamma \exp\left(-\frac{x^2 + y^2}{\lambda^2}\right)$$

The sensitivity of process to systematic and random errors in the parameters is observed in Figures 2, 3 and 4. These plots depict the improvement in the RMS difference from the desired figure. This information provides a quantified measurement of the relative sensitivity to errors in the beam parameters as well as the positioning of the beam with respect to the part. Experiments are planned to verify the simulations and to provide information on the accuracy and stability of the beam parameters and positioning.

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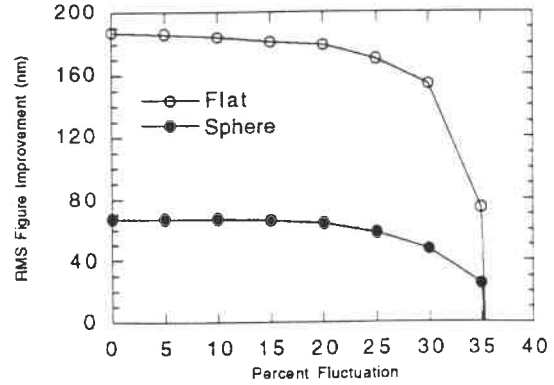
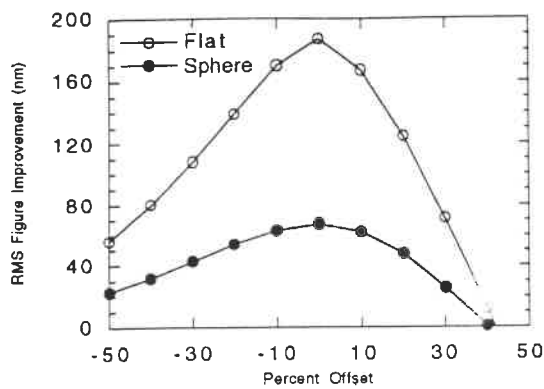


Figure 2 Simulated improvement in RMS figure error shown as a function of systematic and Normally distributed random error as a percentage of the modeled λ .

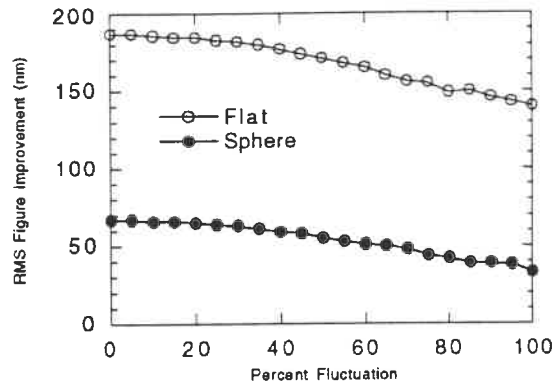
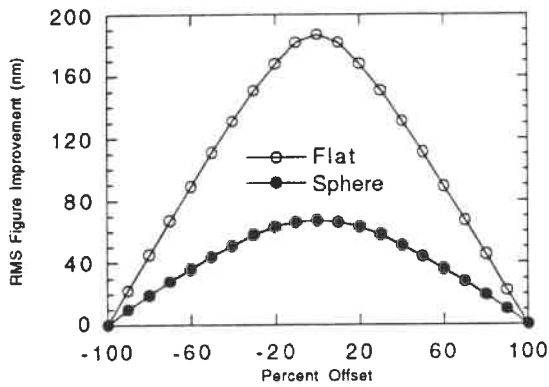


Figure 3 Simulated improvement in RMS figure error shown as a function of systematic and Normally distributed random error as a percentage of the modeled Γ .

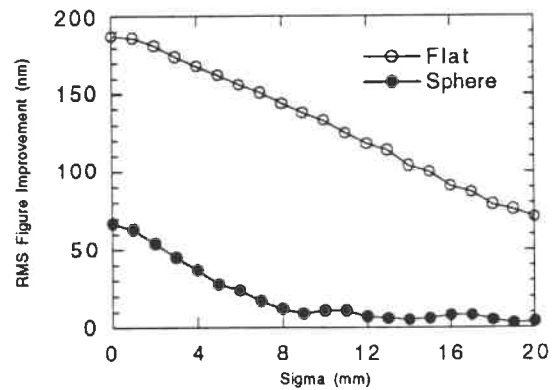
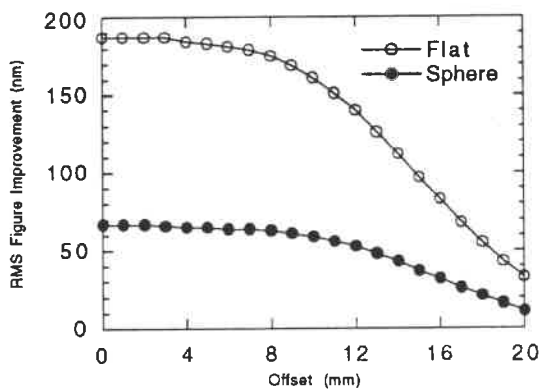


Figure 4 Simulated improvement in RMS figure error shown as a function of systematic and Normally distributed random position error.

Nanoscale Plasticity in Silica Glass

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High precision machining of glass and ceramic surfaces impacts a large number of materials fabrication technologies around the world. The principal difficulty facing many machining techniques is the brittle nature of the materials which results in cracking and other forms of damage. However, there is a growing body of experimental evidence that under certain conditions, usually associated with small depths of cut, even these materials will behave in a ductile manner and material removal can occur with minimal damage. The ability to understand and manipulate this "ductile-brittle" transition is, therefore, critical to improving the economic viability of state-of-the-art machining capabilities.

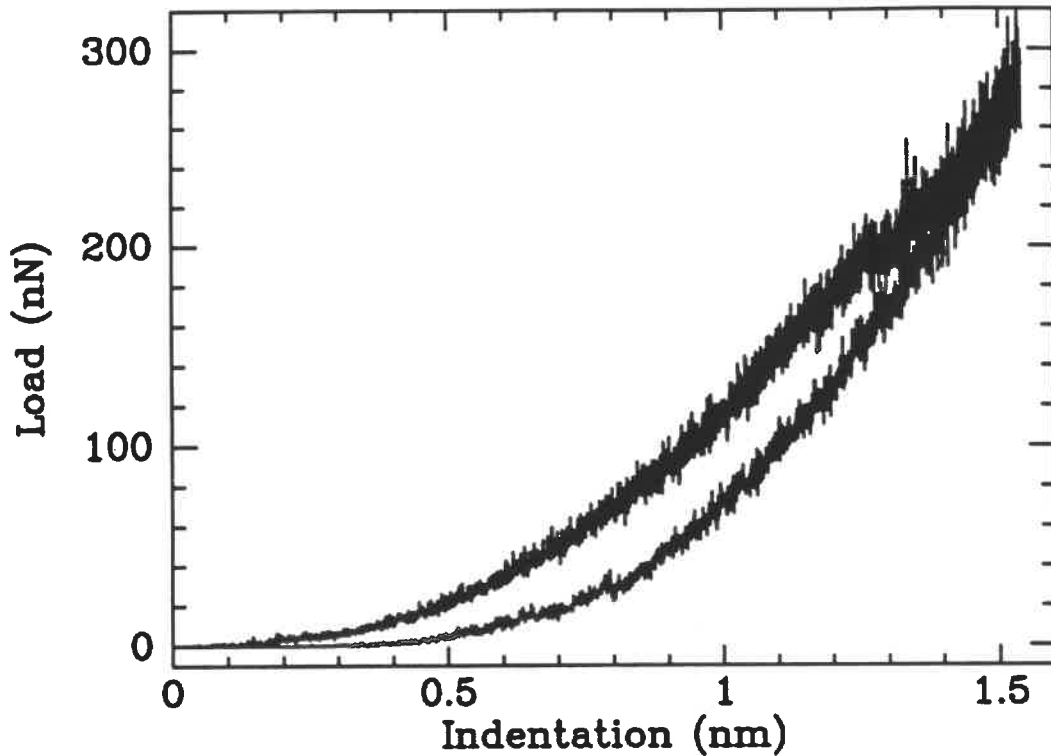
In this paper we examine the mechanisms of nano-scale plasticity and damage initiation in silica glass by using molecular dynamics (MD) simulation techniques.¹ Computer experiments are carried out by indenting a sharp diamond-like tool, containing 4496 atoms, into a silica slab consisting of 12288 atoms. Application of periodic boundary conditions in the (x,y) plane allows the slab to represent a small piece of an extended surface. In order to have a free surface at the top of the slab, no periodicity is applied in the z-direction, and most of the atoms in the silica surface are allowed to move freely according to Newton's laws. To simulate thermal contact with the bulk material below the slab, a relatively few atoms near the lower boundary of the calculation have additional constraint forces which maintain their temperature at about 300° K, while the lowest layer of atoms are held rigidly in place. Because the motion of each atom is followed in detail, changes in the structure of the atomistic, glass network are monitored throughout the calculation.

The nature of the interatomic forces in silica is beginning to be understood, and both purely covalent² and ionic³ potential models have been proposed. Both types of models give comparable agreement with experimental equilibrium structural properties of glass. In this work we use the covalent potential model proposed by Stixrude and Bukowinski,² which contains four terms to represent Si-O bond-stretching, O-Si-O and Si-O-Si angle bending, and O-O repulsion. This potential accurately reproduces the experimental structure, as represented by neutron and x-ray scattering data, as well as the elastic properties of α -quartz and SiO₂ glass, and it should give a reasonable representation of a silica surface.

The atoms in the indenting tool are assumed to maintain a rigid diamond structure that

moves with a constant velocity perpendicular to the surface. The indenter is triangular and is formed by chopping the corner off of a cube and exposing a $[111]$ plane, which forms the bottom of the indenter. The sides of the indenter are the corresponding $[100]$ planes, which formed the corner of the original cube. The interaction of the “diamond” atoms with the individual Si and O atoms of the surface is represented by the repulsive part of a Lennard-Jones potential.

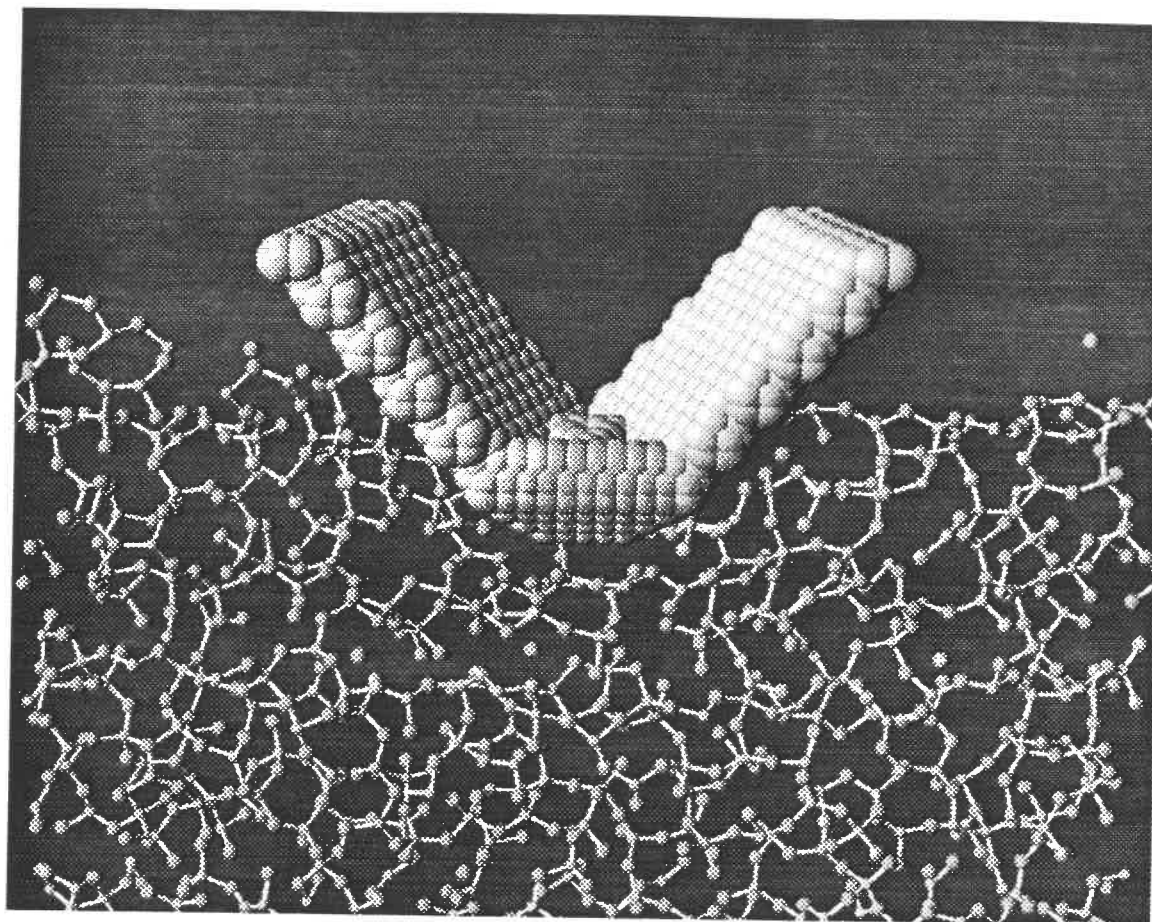
FIG. 1. The load on the tool as a function of indentation. The upper curve is the load during indentation and the lower curve is the load during removal. The kink at an indentation of 1.25nm corresponds to the onset of plasticity.



To create the initial slab, we reproduced a configuration of 192 atoms from a bulk simulation of silica⁴ in a $4 \times 4 \times 4$ array. The resulting large cell of atoms was “cooked” at a very high temperature and then quenched to a moderate temperature with periodic boundary conditions applied in all three directions. At about 1000° K the periodicity in the vertical direction was removed and the surface was quenched further until it reached 300° K. When the periodicity in the z -direction was removed, the slab expanded somewhat in that direction, and the top of the slab was observed to oscillate up and down. Care was taken to

assure that the expansion was complete and that the oscillations were damped away before any indentations were started.

FIG. 2. A cross-sectional slice of width 0.6 nm through the center of the simulation cell at an indentation of 1.5 nm. Silica glass forms a random network of interconnected tetrahedra. The onset of plasticity is signalled by a rearrangement of the network at about 1 nm directly beneath the tip.



In Figure 1 we show the load on the tool as a function of indentation for a simulation at 50 m/s. The curve is reminiscent of recent nanoindentation experiments⁵ though we do not observe any crack formation in our simulations. For indentations beneath 1.25nm the surface responds elastically and the loading curve is reversible upon removing the tool. At an indentation of 1.25nm we observe a kink in the loading curve. At this point in the simulation a significant rearrangement of the tetrahedral network (see Figure 2) occurs directly beneath the tool, accompanied by an increase in the number of silicon-oxygen bonds and a rise in the local temperature. Reversing the tool from beyond this point leads to the hysteresis loop as

shown in Figure 1. The relatively small hysteresis indicates that most of the deformation is accommodated elastically. The contact pressure (hardness) estimated from the load and the observed area of contact for an indentation of 1.5nm is about 15GPa which is in reasonable agreement with micro-indentation studies performed by Bifano.⁶

In summary, we observe both elastic and plastic deformation of silica during nanoindentation simulation. The transition occurs at an indentation of 1.25nm and the calculated hardness agrees with experiment.

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Ion Beam Modification of Silicon Carbide for Improved Machinability

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The mechanical properties of ceramics are more sensitive to surface conditions than any other class of structural materials. For high quality structural ceramics, most failures under applied loads initiate at the surface. The origin of failure is usually a tensile component of the applied stress or the propagation of an pre-existing flaw or both. Consequently, the useful strengths of these materials can often be raised by surface treatments which remove the flaws or reduce their severity or which generate a residual compressive stress layer at the surface (e.g. polishing). Flaw severity can be reduced by increasing the strength, hardness and fracture toughness of the material. One technique for introducing impurities or alloying atoms into the near-surface region in a controlled and reproducible manner is ion implantation. The net effect from the stopping of the injected ion is production of point defects which work to increase the strength by alloy hardening (solid solution or second phase precipitation) and/or radiation damage hardening (defect strengthening). Both processes will contribute to the formation of a residual compressive stress state in the near-surface region. [1]

The machinability of brittle ceramics to precision tolerances ($<1\ \mu\text{m}$) is directly related to the mechanical properties at the surface layer which are oftentimes different than bulk values. There is ample evidence that the transition to ductile behavior in brittle materials can be correlated with a critical scale of machining associated with the fracture toughness, hardness and elastic modulus. Research and development activities at ORNL in low / zero-damage deterministic machining are seeking to exploit these phenomena to replace costly polishing of reflective optics and to enhance the cost-effective machining of ceramics for heat-engine applications. In previous work, the level of fluence induced minor damage in crystalline SiC ranging up to amorphization. Qualitative improvements in the mechanical properties of the implanted layer and nearby substrate were observed. Implantation of silicon ions into a polycrystalline β -SiC should create a surface layer exhibiting amorphous characteristics to depths on the order-of a micron (μm). The results suggest that the material would become more amenable to ductile removal mechanisms. Several specimens of silicon carbide have been implanted with ions injected by an large isochronous generator at ORNL. Experiments were run with three precision fabrication technologies: single point diamond turning, ductile grinding and ion beam milling to validate the machinability improvement. Results from these experiments will be presented and discussed.

Experiment Description

In this initial study, a single silicon carbide material was selected. The samples of chemical vapor deposited silicon carbide (CVD SiC) were obtained from Morton International and are essentially pure β phase (cubic) material. Five disc specimens measuring 40mm in diameter and 3.18 mm thick were polished at General Optics, Inc. using proprietary techniques. Each specimen was received with an average surface finish of approximately 1.5 Å RMS as measured over a 1mm scan with a Chapman MP2000 laser profilometer using a 10x objective and a 0.050 mm bandpass filter. Specimens were also characterized prior to implantation with a BRO OmniScatR scatterometer which provided power spectral density (PSD) information and bidirectional reflectance distribution function (BRDF) light scattering at 10 degrees off specular. These data are presented in Table 1 and suggest that the smooth polished surface was disrupted by erosion of particles and injection of dislocation damage during exposure.

Table 1. Pre- and post-implantation measures of surface roughness on SiC specimens.

Specimen	Surf. Finish (Å RMS)		BRDF (sr ⁻¹)	
	Pre-	Post-	Pre-	Post-
S-095	1.65	11.61	1 E-03	1.7E-03
S-096	1.44	24.65	6.9E-04	2.9E-03
S-097	0.99	21.29	1.5E-03	2 E-03
S-098	1.54	11.03	7.8E-04	2.5E-03
S-099	1.52	16.59	4.5E-03	1.7E-03

Ion implantation was performed at an Oak Ridge National Laboratory (ORNL) Ion Research Users' Center (5MV Van de Graaf). The implantation parameters included Si cations only, energized to 5 MeV and at a fluence of 10¹⁶ ions/cm². The specimens were controlled to constant temperature under ion heating by a liquid nitrogen coolant (77 K).

Experiment Modeling

Implantation simulation was performed using the TRIM software package. The program predicted a mean damage depth of approximately 2 µm. Previous experience with Cr⁺ implantation into SiC, both single 6H crystal and polycrystalline sintered α -SiC, at 280 keV yielded an amorphous layer which extended 250 - 300 nm deep. Additionally, TRIM was utilized to estimate the sputter yield as a measure of surface erosion of material under ion bombardment. TRIM predicted that approximately 300 nm of depth would be removed. This phenomena renders the net apparent implantation damage depth to some lesser number. A Tencor step profilometer was employed to measure the surface topography of the implanted-to-masked interface. Multiple traverses across the interface in several specimens gave an average step of 200 nm.

Experiment Results

Post-implantation tests were conducted on the specimens with two methods and at separate locations. One specimen was subjected to nanoindentation and instrumented diamond stylus scratching tests at ORNL. A second specimen was sent to Univ. of California, Santa Barbara

for higher resolution scratching studies using an atomic force microscope (AFM). The AFM was refitted with a single crystal diamond tip. The ORNL tests included a constant 9 gram load as the stylus was dragged across the interface. Additionally, the stylus was subjected to a ramped load up to a 9 gram maximum and dragged within each region. In both sets of tests, the mechanical properties on the implanted surface appeared to have been dramatically altered. There was a clear propensity for the masked (non-implanted) region to exhibit cracking under constant load, while the implanted region exhibited more ductile flow of material. Similarly, the ramped loading resulted in earlier fractures appearing in the masked region. Preliminary results from normal and tangential force measurements during scratching also suggest that the coefficient of friction was not significantly impacted by the implantation.

Acknowledgment

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Key Words: ion beam, ion implantation, silicon carbide, precision grinding, diamond turning

ION BEAM SHARPENING OF QUADRANGULAR AND TRIANGULAR PYRAMIDAL DIAMOND INDENTERS OR PROBES

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1. Introduction

The scanning tunneling microscope (STM) [1] has become a powerful tool for studying the micro-topographical and electrical surface structures of materials in atomic resolutions. Moreover the STM can be used to fabricate structures in the nanometer (nm) scale by electron beam assisted photo-chemical etching and electron beam assisted chemical vapor deposition, and to create nano-structures by direct indentation with mechanical manner. Usually, a tungsten or platinum tip fabricated by electro-chemical polishing [2] and mechanical grinding is usually used as a probe. It is, however, difficult to define the profile of a tip made of tungsten and platinum in nm dimensions by above mentioned methods and to keep the sharpness of the tip during and after nano-indentation.

In contrast to those probes and processing methods, the profile of a diamond indenter or probe in nm scale can be easily defined by the ion beam machining [3] combined with the mechanical lapping. Pure diamond with an energy gap of 5.5 eV, however, is an excellent electric insulator at the room temperature. Therefore, to increase the electrical conductivity of indenters, we used boron doped single crystal diamond tips which were made artificially. The tips were pre-finished by mechanical lapping, and have truncated right quadrangular and triangular pyramidal shapes. In this study, sharpening principles of the above mentioned probes were mainly investigated.

2. Experimental apparatus and procedure

Truncated quadrangular and triangular pyramidal shape diamond indenters or probes pre-finished by mechanical lapping were sharpened using the ion shower type machining apparatus equipped with a Kaufman type ion source. The apparatus can generate argon ion beams of an ion energy $E=1.0$ keV at an ion current density $i=0.3-0.5$ mA/cm².

The profile and surface topography of the indenters or probes were observed with a scanning electron microscope with four secondary electron detectors [4].

A HOPG (Highly Ordered Pyrolytic Graphite) sample, which is the standard specimen for the STM measurement, were observed with a commercially available STM using the processed diamond indenters or probes. The STM has a vibration isolator of five stacked metal plates with Viton spacers, a cross type piezo actuator for x-y scanning and z-servo., and a inchworm.

3. Sharpening principle of indenters

(a) Truncated right quadrangular pyramidal indenters

Sharpening of a truncated quadrangular diamond indenter by ion beam machining with ion incidences of vertical (V direction) or horizontal (H direction) to the indenter is illustrated in Fig.1(a), which represents a three-dimensional shape of the indenter. As shown in the figure, the indenter having initial tip width of X_0 and Y_0 ($Y_0 > X_0$) will be sharpened by ion beam machining, because four inclined planes are machined faster than the top plane. After ion beam machining of the indenter with vertical ion incidence for t hours, the tip width X_t and Y_t is given by the equation (1). However, after $X_t = 0$, X_t and Y_t is governed by equation (2). Namely, we can not sharpen the indenter.

$$\begin{aligned} \text{Before } X_t = 0: \quad X_t &= X_0 - \frac{2}{\tan \alpha} \frac{V_n(\alpha)}{\cos \alpha} - V_n(0) \cdot t \\ Y_t &= Y_0 - \frac{2}{\tan \beta} \frac{V_n(\beta)}{\cos \beta} - V_n(0) \cdot t \end{aligned} \quad \text{----(1)}$$

$$\text{after } X_t = 0: \quad X_t = 0 \quad \text{and} \quad Y_t = Y_0 - X_0 \quad \text{-----(2)}$$

where, $V_n(\alpha)$, $V_n(\beta)$ and $V_n(0)$ are ion beam machining rate of the plane having inclination angle α , β and 0 , respectively.

If a truncated quadrangular diamond indenter having knife edge tip will be machined with horizontal ion incidence, then the indenter will be sharpened. However, it is very difficult to control the machining time for just sharpening.

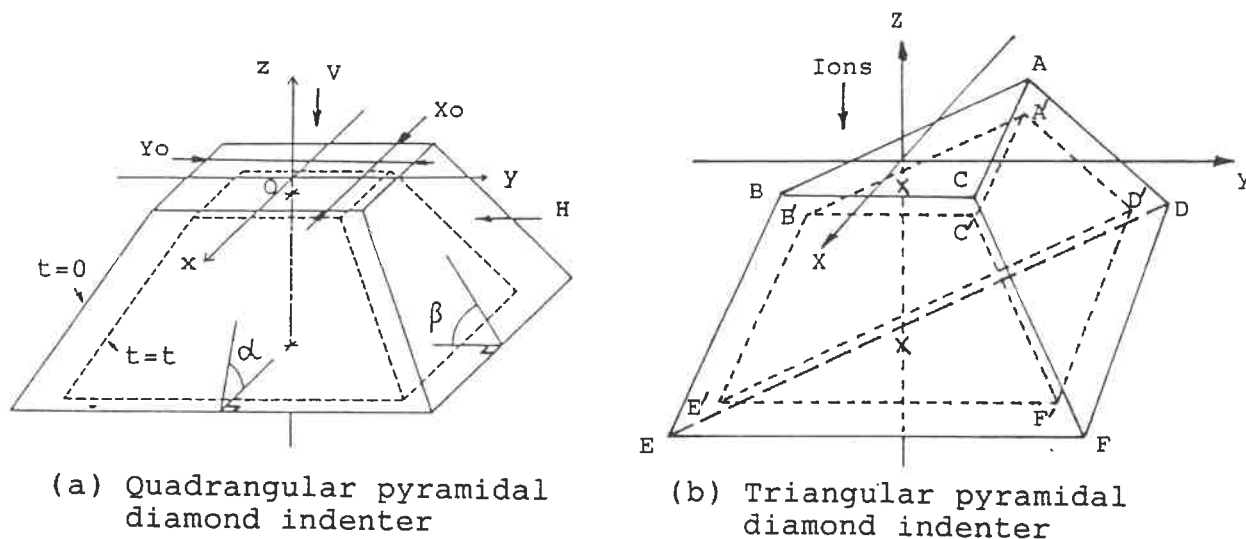


Fig.1 Three dimensional model of indentors for predicting the length of the sides of the top surface during ion beam machining

(b) Truncated right triangular pyramidal indenters

Sharpening of a truncated right triangular pyramidal indenter by ion beam machining at the vertical ion incidence to the top surface is illustrated in Fig.1(b). As shown in the figure, three sides $\overline{AB}$, $\overline{BC}$ and $\overline{CA}$ of the triangle ABC which is the top surface of the indenter move parallel to themselves during ion beam machining, therefore the triangle A'B'C' at the top surface of the indenter machined for t hours becomes similar figures of the triangle ABC. Then, the following equation is obtained:

$$\overline{AB}/\overline{A'B'} = \overline{BC}/\overline{B'C'} = \overline{CA}/\overline{C'A'} \text{------(3).}$$

Here, we introduce the decreasing rate Vs of the side $\overline{AB}$, then the side $\overline{A'B'}$ of the indenter machined for t hours is expressed with equation (4).

$$\overline{A'B'} = \overline{AB} - V_s \cdot t \text{------(4).}$$

From equations (3) and (4), the following equations are reduced:

$$\overline{B'C'} = (\overline{BC}/\overline{AB})(\overline{AB} - V_s \cdot t) \text{------(5)}$$

$$\overline{C'A'} = (\overline{CA}/\overline{AB})(\overline{AB} - V_s \cdot t) \text{------(6).}$$

From equations (4), (5) and (6), three sides $\overline{A'B'}$, $\overline{C'B'}$ and $\overline{C'A'}$ of the top surface become zero after machining for $\overline{AB}/V_s$ hours.

3. Results and discussion

Fig.2 shows the machining time dependence of the tip width X_t and Y_t of a truncated quadrangular pyramidal indenter pre-finished by mechanical lapping and machined with 1.0 keV argon ions at vertical ion incidence (V direction) for 7 hours. As shown in the figure, the tip in one direction was sharpened. However, the indenter still has the knife edge shaped. Namely, after $X_t=0$, Y_t becomes Y_0-X_0 following the equation (2). Fig.3 shows the SEM photos of a truncated triangular pyramidal indenter machined with 1.0 keV Ar ions at vertical ion incidence to the top surface for 1 hour. As shown in the

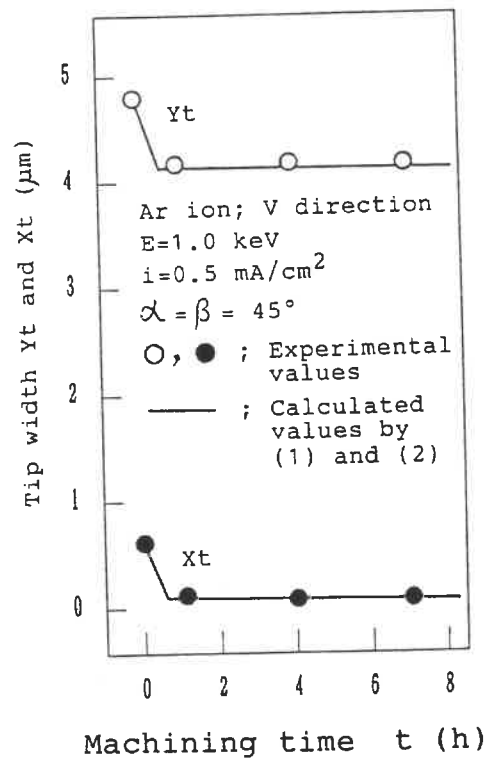


Fig.2 Machining time dependence of tip width Y_t and X_t

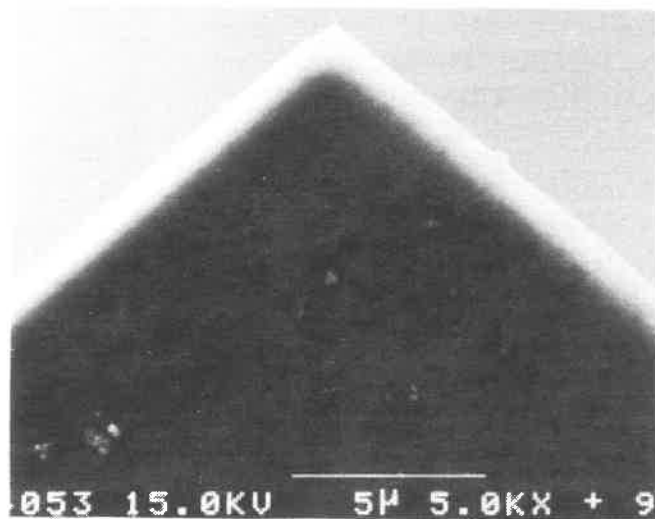


Fig.3 SEM photo of a sharpened triangular pyramidal diamond indenter or probe

Xspan: 2.0 nm
Yspan: 2.0 nm
Zp-p : 0.3 nm

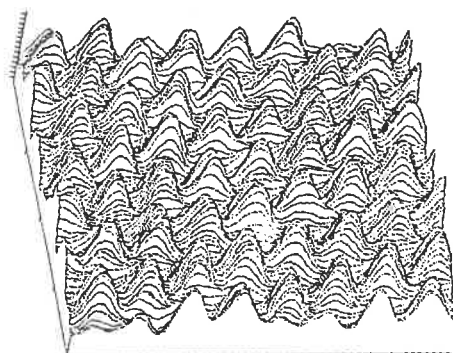


Fig.4 STM image of HOPG taken with a diamond probe
($I_t=1$ nA, $V_b=50$ mV)

figure, we can always sharpen the trigangular pyramidal indenter. The STM images of HOPG, obtained with the diamond probe or indenter, are shown in Fig.4. As shown in the figure, atomic STM images of HOPG were obtained using semiconductor diamond probe in spite of the wide tip width of y direction. Atomic STM image may be obtained with a sharp edge of the inclined indenter.

4. Conclusion

From the experiments, conclusions are summarized as follows:

- (1) Tip widths of truncated quadangular and triangular pyramidal diamond probes or indentors machined with 1.0 keV argon ions at the vertical ion incidence can be predicted with simple equations.
- (2) If a truncated right quadangular pyramidal diamond probe has not a square top surface, the probes can not be sharpened by ion beam machining at the vertical ion incidence.
- (3) A truncated triangular pyramidal probe can be sharpened in spite of its shape.
- (4) The boron doped diamond probes with triangular and quadangular pyramidal shape can be used to indent and to imege the surface of HOPG (highly ordered pyrolitic graphite) in ambient air.

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Development of optical fiber connector polishing machines and ellipsometric characterization of the polished surface damage

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Optical fiber communication systems use many optical connectors, and signal quality in these systems -especially in analog CATV transmission system- is extremely sensitive to reflection occurring at connector end surface, where optical reflection causes noise resulting from the light returned to the laser diode and from multi-reflection between connectors [1]. Because the polishing of the end surface of optical fiber is therefore very crucial, this presentation introduces inexpensive, high-performance, and easy-to-use polishing machines for optical connector end surface. We also describe and demonstrate the effectiveness of an ellipsometric method for quantitatively characterizing the damage on the polished surface.

(1) High-performance polishing machines

We have developed two kinds of polishing machines: a mass-production machine to reduce the cost of polishing; and a hand-held machine for easy use in the field. In this polishing, the surface shape precision (convex curvature radius, fiber eccentricity, and height difference between a fiber and a ferrule), optical characteristics (return loss), and polishing cost must be satisfied. The principle by which they form a convex shape is shown in Fig. 1. Both machines use elastic rubber plates with plastic films and polishing slurry to form convex shape end surfaces, and both can polish the fibers in three steps: adhesive removal, rough polishing, and final polishing. The polishing slurry for both machines are diamond and SiO_2 .

The mass-production machine can polish up to 12 fibers at the same time, and the polishing plate makes revolution and rotation movement. The total polishing time is 5 minutes, and the consuming cost per fiber is low, because all the fibers are polished at the same time. Therefore, the machine can decrease polishing cost by more than 50% compared with the conventional one [2].

The hand-held machine polishes a single fiber with a pre-formed ferrule, and the polishing principle is the same as the mass-production one. The weight of the hand-held machine (less than 1 kg) is only one-tenth that of the conventional polishing machine, and is suitable for field use. The machine can polish not only a ferrule with a fiber but also an assembled connector, so it is possible to improve the return loss of the conventional connector, for example, from 25 dB to more than 40 dB.

The characteristics of surfaces polished by either machine are adequate for use in communication systems: the return loss is more than 40 dB, the surface shape satisfies specifications; eccentricity is within 50 μm , convex curvature radius is 10-25 mm, and the height difference between a fiber and a ferrule is 0.05 μm or less (Fig. 2).

Photographs of the developed machines are shown in Fig. 3.

(2) Characterization of surface damage due to polishing

Polishing-induced damage affects the optical characteristics of a fiber, especially return loss. To quantitatively characterize the induced damage, we ellipsometrically evaluated the refractive index n and the thickness d of the damaged layer under the polished surface and then compared these results with the return loss. Typical specimens in this experiment were optical fibers polished by the diamond lapping film with diameters of 0.5-3 μm diamond abrasive. These specimens were characterized by ellipsometry, optical return loss measurement, and measurement with an atomic force micrometer (AFM). The solid lines in Fig. 4 show the calculated return loss, and the circles show the ellipsometrically evaluated n and d values. It has become clear that damaged layers caused by polishing using diamond abrasives have rather large optical indexes (about 1.54) in comparison with the original value (1.46, at a wavelength of 632.8 nm). It is also clear that thickness of the damaged layer decreases with decreasing diameter of the abrasive. The return losses also improve as the diameter of the abrasive decreases. The increase in the index is due to the density increase in the damaged layer caused by the residual strain of processing [3]. The close correspondence between the measured return loss and the return loss calculated from the ellipsometrically evaluated n and d values (Fig. 5) confirms that this nondestructive, quantitative, and easy-to-use ellipsometric method can accurately characterize the surface damage. The return loss and results of ellipsometric characterization were also compared with the results of an AFM measurement. As shown in Fig. 6, there are microscratches on the surfaces, and the scratches become shallower as the abrasive diameter decreases, corresponding to the reduction in the thickness of the damaged layer. For each surface, the depth of scratch is ten times less than the ellipsometrically evaluated thickness of the damaged layer. These characterization results thus also correspond with the results of an AFM measurement.

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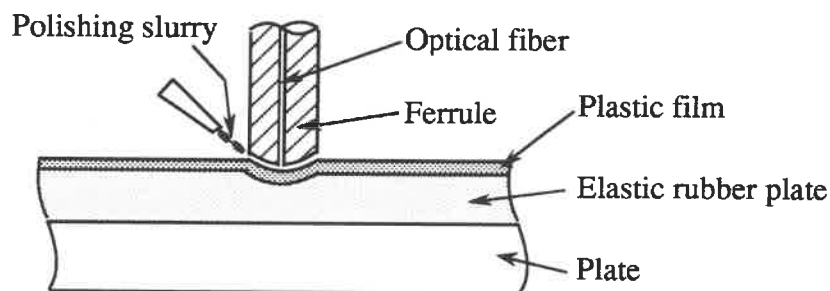


Fig. 1 Principle of the polishing using an elastic rubber plate with a plastic film and polishing slurry to form a convex shape for optical fiber end.

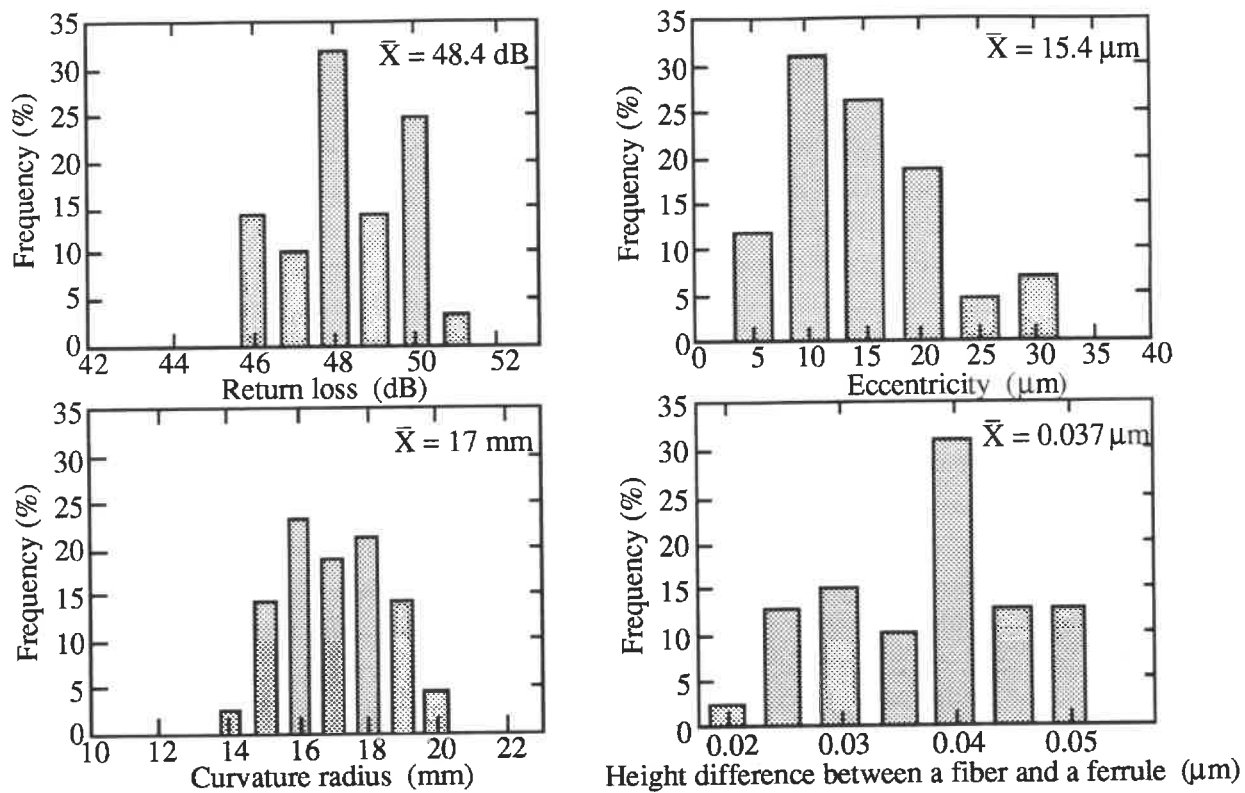
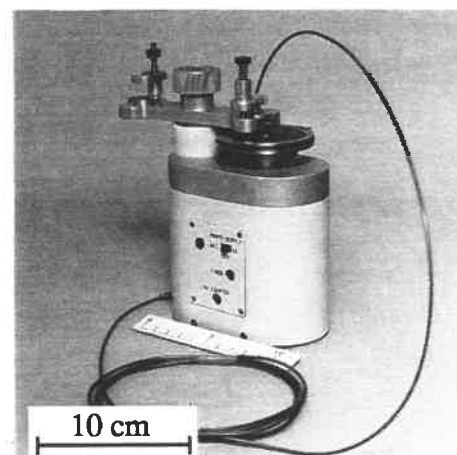


Fig. 2 Optical characteristics (return loss) and surface shape precision of the polished surfaces by the mass-production machine.



(a) mass-production machine



(b) hand-held machine

Fig. 3 Developed polishing machines.

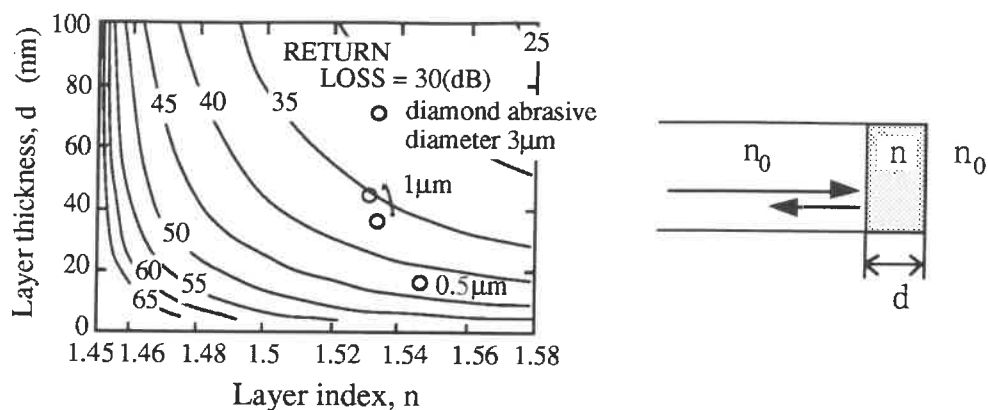


Fig. 4 Thickness and refractive index of the surface layer damaged by polishing (lines show the return loss calculated under the assumption of one damaged layer structure, and circles show the ellipsometrically evaluated values).

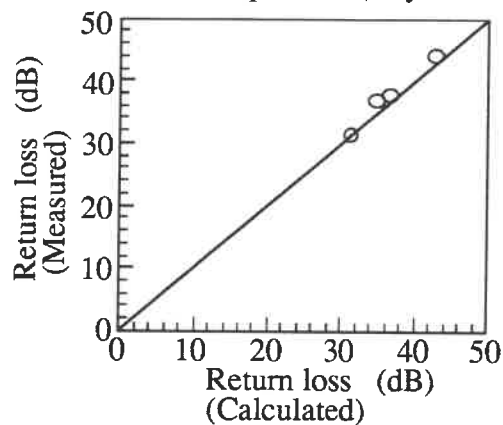
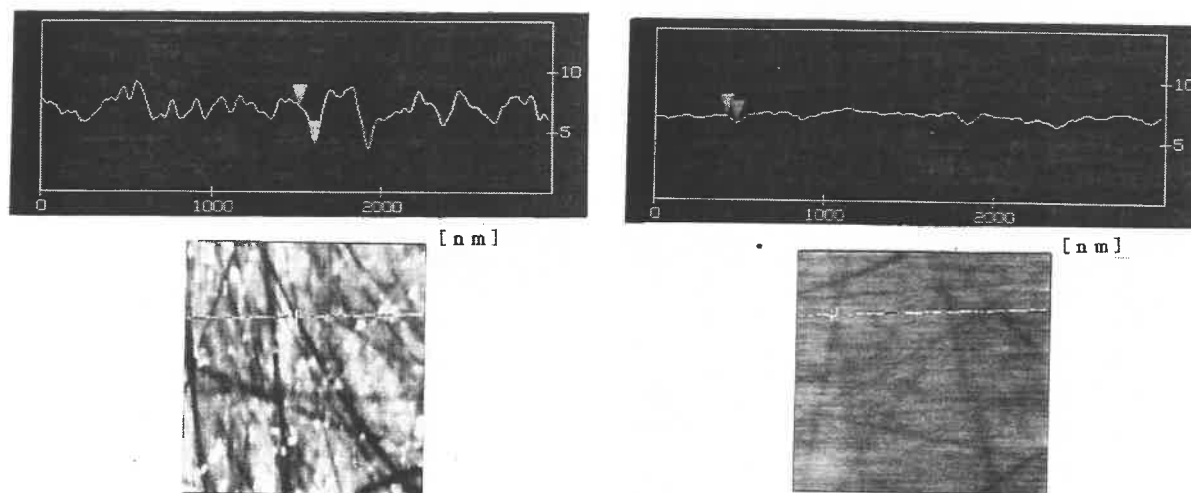


Fig. 5 Correspondence of the calculated and measured return loss.



(a) 1 μm diameter diamond abrasive

(b) 0.5 μm diameter diamond abrasive

Fig. 6 Polished surfaces measured by AFM.

Precision Machining and Measurement Organized by Miniature Robots

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1.Introduction

A new trial approach to precision machining, measuring and transporting organized by several miniature robots which are capable of moving and working on the workpiece precisely is presented. In conventional machine tools and measuring machines, several techniques for improving the spatial positioning accuracy have been used while these machines tend to be suffer from environmental conditions such as temperature fluctuation[1] and floor vibration when discussing the stability of less than sub-micron. We have the experiences that the scanning probe microscopes have an atomic or a molecular size level resolution enough to investigate microscopic topography[2]. This means that miniaturized machine structure has the higher mechanical stability which is advantageous to ultra precision region. It is obvious, however, that the small machine has poor dynamic working range, typically a few micrometer in the STM instruments. So it is also demanded to expand the working range of such small machine as well as the applicability to actual precision engineering tasks. **Figure 1** shows the conceptual image where many precision miniature robots, controllers and transducers are distributed on the workpiece and organized to pursue their precision tasks[3]. Consequently it is expected that miniature robots should collaborate with the conventional machines for cost saving benefit and wider mobility.

We describes several miniature robots which have made to move precisely on not only the horizontal plane but also a vertical plane including curved surface. And it is demonstrated that some of them have the ability of fine single point cutting, roughness measurement and simple dimensional measurement as a trial application.

2.Mechanism of Precision Mini-Walker

2.1 Basic Components

We designed and fabricated miniature walking machines which consist of piezo elements for thrusting and electromagnets for clamping as shown in **Fig.2**[4]. The small machine, as small as a golf ball, is basically driven by synchronizing the magnetic attraction and the expansion/contraction of piezo element such as an inchworm mechanism. Two U-shaped small magnetic legs are mechanically connected with piezo elements through a flexure joint or a special bearing washer. This arrangement allows it to move on any inclined curved surfaces even a wall and a ceiling although the target surface is limited to ferromagnetic materials. We should design carefully the specification of electromagnet leg as well as its magnetic characteristics. And electrical current must be controlled critically so that magnetic force could be strong enough for clamping against the machine weight and the thrust force from piezo element. But higher electrical current is never supplied because heat generation should cause significant damage to the machine.

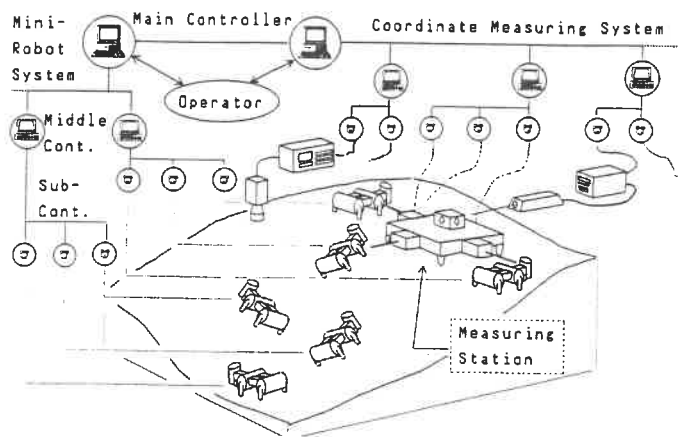


Fig1. Network of miniature robots, controllers and transducers

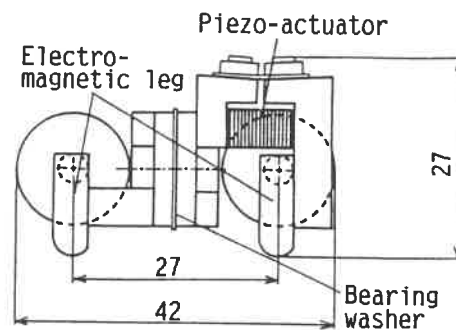


Fig.2 Miniature walker with piezo elements and electromagnetic legs.

2.2 Performance

We show the experimental results of measured moving leg response to the control signals in Fig.3 and the photograph of miniature machine climbing up S-curved surface in Fig.4. We achieved successfully the good performances that the small machine can move up and down on the curved surface with $20\mu\text{m}$ step width at a speed of 2mm/s .

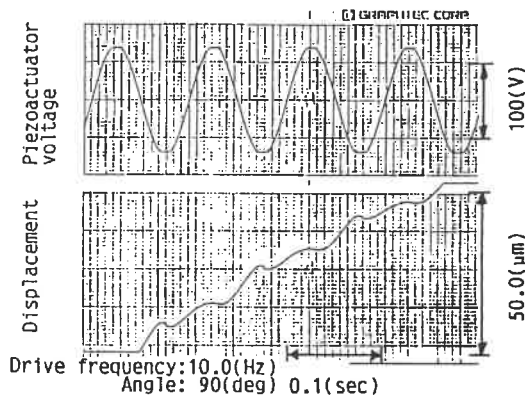


Fig.3 Dynamic displacement of miniature walker

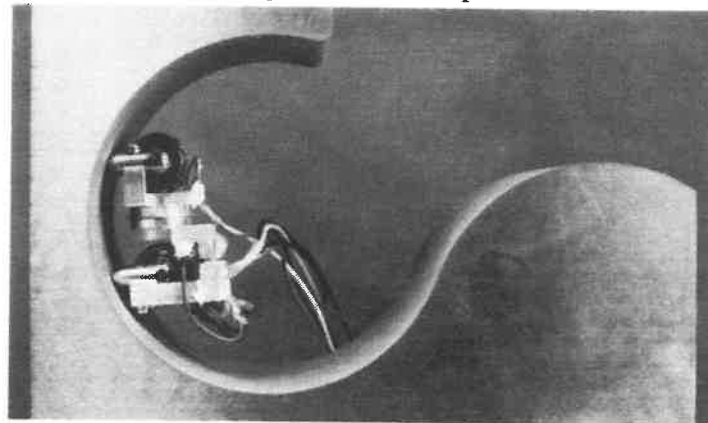


Fig.4 Photograph of mini-walker which is climbing up S-curved surface.

3.Application to Precision Tasks

3.1 Single Point Fine Cutting

We also made a miniature robot capable of surface fine cutting as shown in Fig.5. Small machining unit composed of a micro-stepping motor with a crank axis, a flexure arm with a diamond single tool at the endpoint is mounted on the "micro-cart" pushed by the "micro-tractor". These components are carefully assembled so that the motor rotation can provide the circular tool path and the cutting force can be also adjustable. By synchronizing the motor rotation with the micro-cart translation, we can obtain scratched lines which are spaced precisely as a ruling engine makes a grating. Figure 6 shows the grooves machined on a glass sample comparing with the reference of $10\mu\text{m}$ scale. We can achieved several grooves spacing approximately $7.5\mu\text{m}$, attaching the tool of $5\mu\text{m}$ radius. We are currently making efforts toward to improve accuracy and resolution of the mini-robot by using closed-loop control with precision displacement transducer from another one.

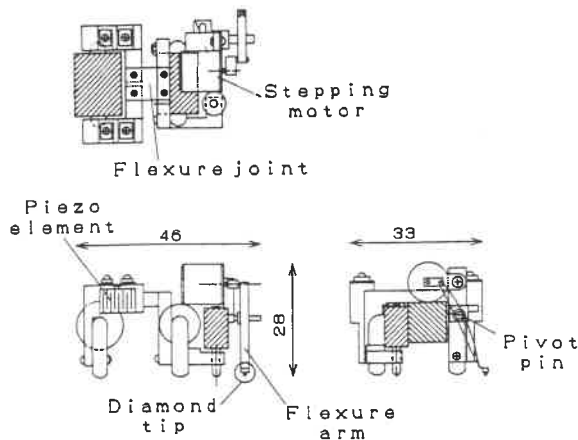


Fig.5 Mini-robot with a single diamond tool

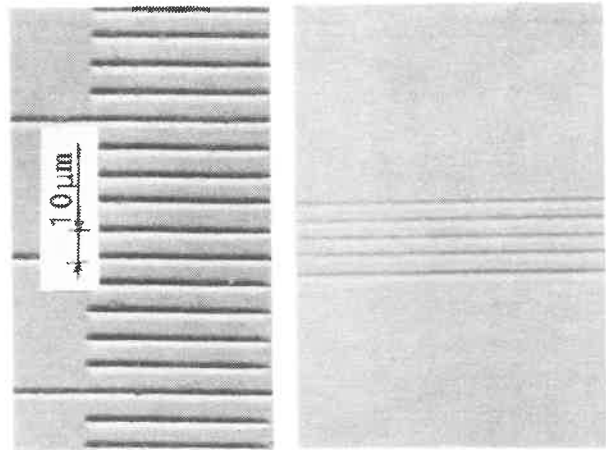


Fig.6 Grooves machined on a glass sample

3.2 Surface roughness measurement

Another mini-robot which is capable of measuring surface roughness is shown in Fig.7. It has a stylus on the cantilever whose bend is simply detected on the photo sensor as a light beam deflection from LED optics. And coordinate position of the mini-robot is monitored by using multi-dimensional motion analyzer(Image Tracker MVA2080) with CCD camera and telescope lens, and then measured data are processed and communicated to a host computer through GP-IB interface. There local sampled data of height changes at the mini-robot and the global information from the visual coordinate measuring system are combined for producing roughness curve as shown in Fig.8.

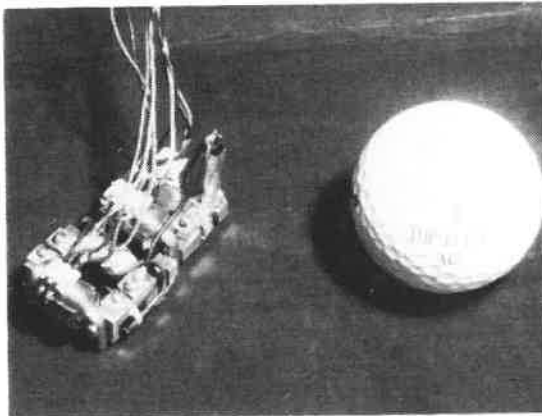


Fig.7 Mini-robot for surface roughness measurement

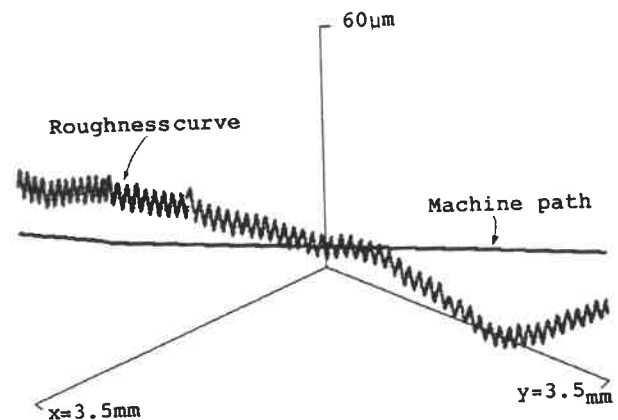


Fig.8 Roughness curve with mini-robot's trajectory

3.3 Simple Dimensional Measurement

Furthermore miniature robots with touch probe have been designed and fabricated for applying dimensional measurement shown in the photograph of Fig. 8, while all operation and control are performed manually yet. Here three mini-robots can be navigated to approach the cylindrical workpiece. The moment that they come into contact with the target, the coordinate positions of the center of three LED can be measured by the visual CCD camera system. We can obtain the diameter of the target when calculating by the responsible computer. Figure 9 is the experimental result of diameter measurement which compares the "Walking Probe Measurement" with the conventional measuring tool.

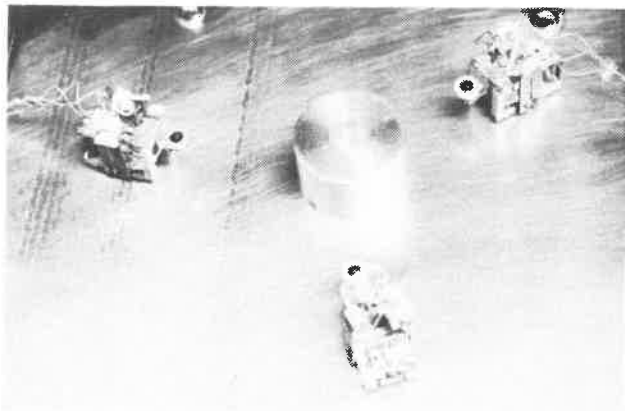


Fig.8 "Walking Touch Probe" with LED

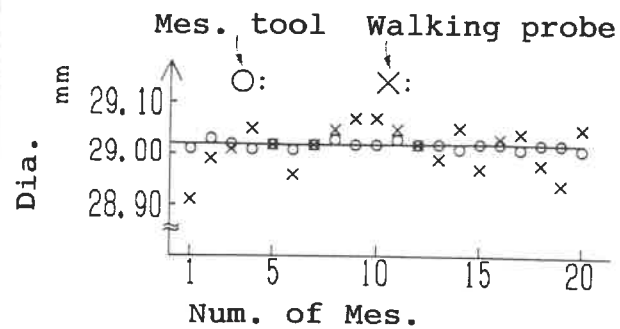


Fig.9 Result of diameter measurement

4. Current Status and Future Work

In this paper, we presented the mechanism of miniature robots which is composed of piezo element and electromagnet for moving precisely. And several performances of surface fine cutting, roughness measurement and simple dimensional measurement were demonstrated. Currently we are designing and fabricating other miniature robots capable of picking up and transporting small parts, manipulating and vacuum-clearing and so on. On the other hand, the framework of organizing the overall system also is considered as shown in Fig.10. We endeavor to organize the hierarchical distributed system involving multiple mini-robots, computers and transducers. It also becomes very important to study the task recognition, decomposition and assignment which are based on domain dependent knowledge so that our miniature robots could perform precision works through a good interface with the operator.

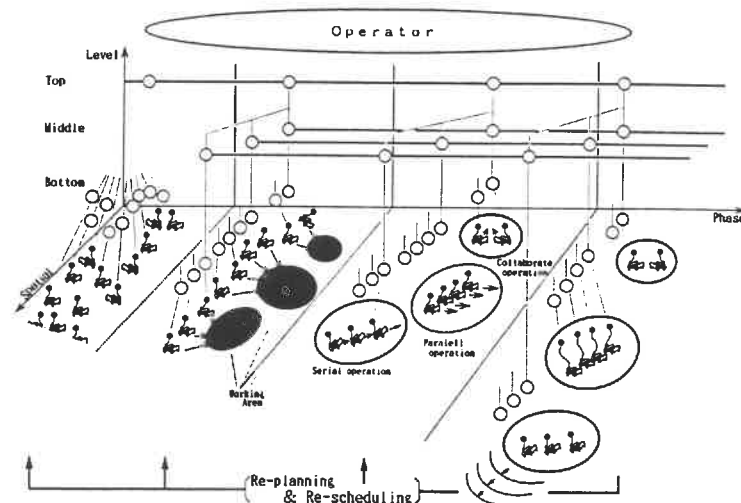


Fig.10 Hierarchical distributed system for organizing miniature robots

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Application of Advanced Ceramics Materials to Sliding Guideways (Friction and Wear Characteristics under Fluid and Solid lubrication)

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Abstract

Advanced ceramic materials which have excellent properties of wear resistance, heat resistance and corrosive resistance were applied to the sliding guideways of machine tools. The friction test equipment used to evaluate the friction and wear characteristics under similar sliding conditions of practical guideways is described. The friction and wear characteristics of ceramic sliding guideways were investigated under fluid and solid lubrication.

1. Introduction

The performance of the guideways is one of the important factors which affect the accuracy and productivity of machine tools. Sliding guideways have been conventionally applied to machine tool tables because of their high damping abilities. Compared to rolling guideways and hydrostatic guideways, however, sliding guideways show some demerits such as stick-slip motion at low sliding velocities and extraordinary wear at high speed operation. An effective method to improve the performance of sliding guideways is to apply advanced ceramic materials to the guideways instead of the usual metallic materials.

This study proposes the application of advanced ceramics to the sliding guideways of machine tools. As the fundamental stage for developing ceramic sliding guideways, the friction and wear characteristics were experimentally analyzed.

2. Experimental equipment and procedure

In the case of sliding guideways, the contact condition between the slider and guideway is classified as a plane-on-plane contact with assumed reciprocal sliding motion. In order to evaluate the friction and wear characteristics under similar conditions to the sliding guideways, a special test equipment was prepared. Friction characteristics were investigated for different ceramic materials and lubricants.

The friction and wear test equipment is illustrated in Fig.1. Two ceramic sliders are installed in the slider holder. The contact pressure between the two sliders and the rail is adjusted by the supply pressure to the air cylinder. Reciprocating motion of the slider holder is

provided by the ball screw and DC servo motor. The frictional force between the two sliders and the rail is measured by strain gauges. Water and water-soluble grinding fluid are supplied as the lubricants, and several others were also tested. In all cases, experiments were carried out after the sliders and rail surfaces were washed with acetone. Initial surface roughness was measured before each friction test. The ceramic surfaces were lapped.

3. Experimental results

3.1 Friction characteristics under fluid lubrication

Fig.2 shows the relationship between the contact pressure and the friction coefficient under water lubrication. Al_2O_3 is applied to the rail. The quite low friction coefficient of less than 0.001 was obtained in almost all ranges of contact pressure when Si_3N_4 was applied to the slider. Fig.3 shows the relationship between the contact pressure and the friction coefficient under grinding fluid lubrication. The same friction characteristics as water lubrication were obtained when Si_3N_4 was applied to the slider. However, An increase of friction coefficient was observed when Al_2O_3 was applied to the slider. The results of the durability test are shown in Fig.4. In both water and grinding fluid lubrication, the friction coefficient was kept to the extremely low level of less than 0.0015 over the entire sliding distance. The stick-slip motion, however, occurred at low sliding velocity and high contact pressure under water lubrication when Al_2O_3 was applied to the slider. This was, fortunately, suppressed by applying grinding fluid.

3.2 Friction characteristics under solid lubrication

Molybdenum disulfide (MoS_2) powder, MoS_2 paste (oil blended) and polytetrafluoroethylene (PTFE) spray (oil blended) were also tested as solid lubricants.

The results are summarized in Fig.5, Fig.6 and Fig.7. In the case of MoS_2 -powder lubrication, the friction coefficient was maintained at almost a constant level regardless of the contact pressure. However, the friction resistance is too high for practical applications. In the case of MoS_2 -paste lubrication, the frictional coefficient decreases as contact pressure increased. The friction characteristics showed the same tendency under PTFE-spray lubrication. and the lowest friction coefficient was obtained among three different solid lubricants applied in these tests. The results of the durability tests are shown in Fig.8. The relatively high friction coefficient was observed under MoS_2 -powder lubrication, and the friction force varies frequently with wide amplitude. This was comparatively well stabilized by applying MoS_2 -paste. PTFE-spray gave quite low friction coefficient and excellent wear-resistive characteristics. Stick-slip motion did not occur in any range of sliding velocities or for any solid lubrication tested.

4. Conclusions

As the fundamental stage for developing the ceramic sliding guideways, the friction and wear characteristics of ceramic materials were investigated under fluid and solid lubrication. The results obtained in this study are summarized as follows;

- (1) Quite excellent friction and wear characteristics were obtained using fluid lubricants. Especially, the Si₃N₄ slider gave extremely low friction coefficient and excellent wear-resistive characteristics.
- (2) Grinding fluid can be used as a lubricant when ceramic guideways are applied to grinding machines.
- (3) MoS₂ powder is not suitable for practical application.
- (4) There is a possibility that PTFE-spray can be applied to the lubricant because it shows excellent friction and wear-resistive characteristics.
- (5) Stick-slip motion did not occur with any solid lubricant tested.

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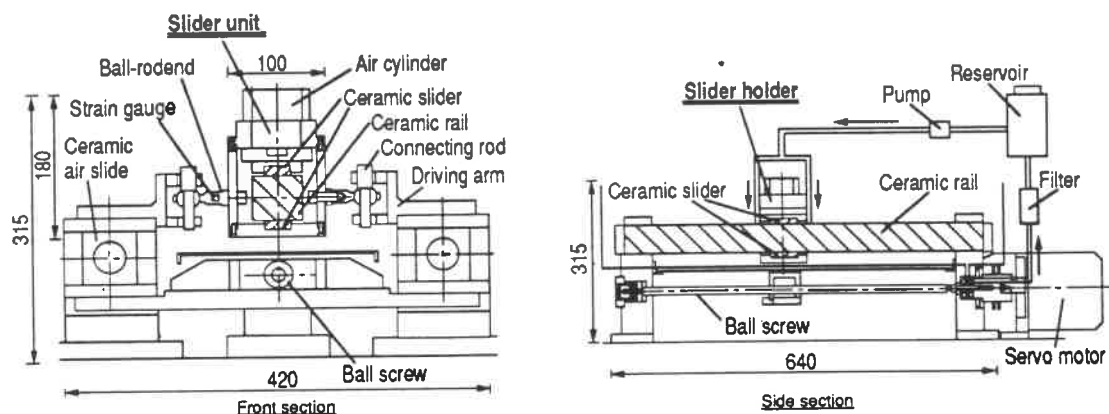


Fig.1 Illustration of the friction and wear test equipment

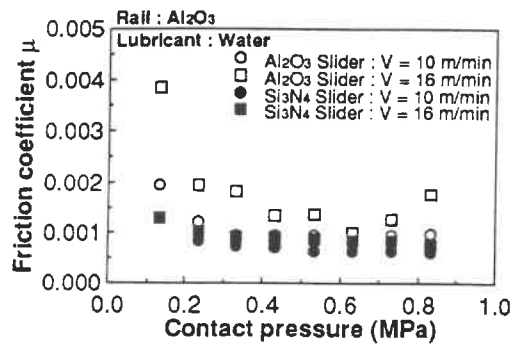


Fig. 2 Friction characteristics under water lubrication

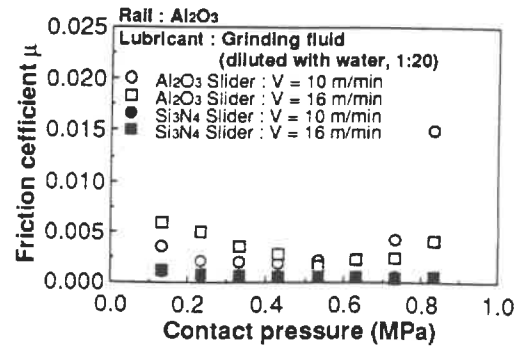


Fig. 3 Friction characteristics under grinding fluid lubrication

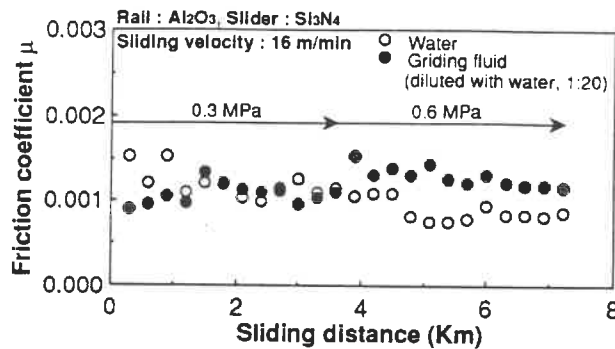


Fig. 4 Results of the durability test under fluid lubrication

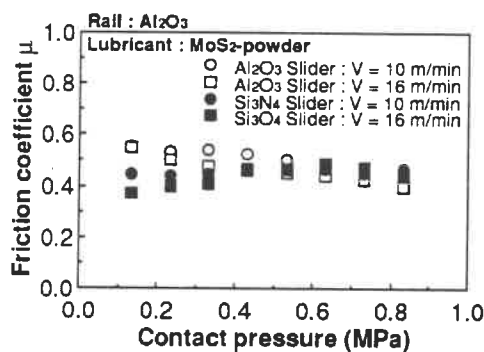


Fig. 5 Friction characteristics under MoS₂-powder lubrication

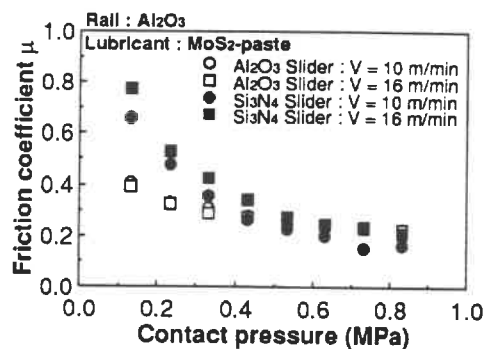


Fig. 6 Friction characteristics under MoS₂-paste lubrication

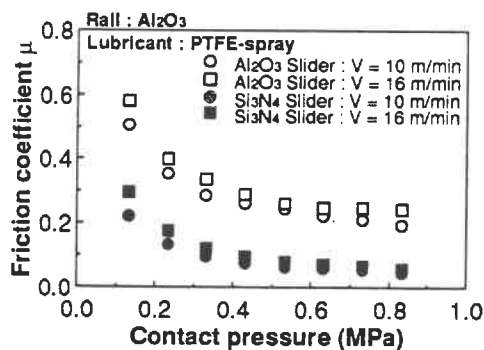


Fig. 7 Friction characteristics under PTFE-spray lubrication

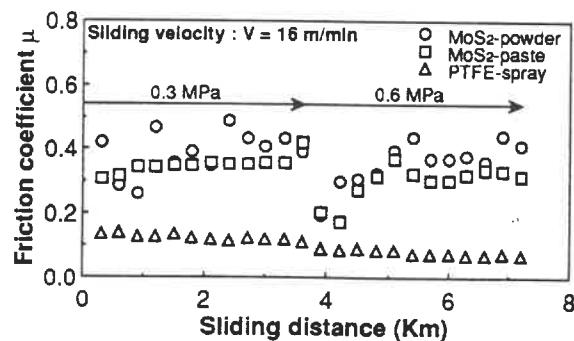


Fig. 8 Results of the durability tests under solid lubrication

COMPENSATION FOR NONLINEAR EFFECTS IN PIEZOELECTRIC ACTUATORS

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SUMMARY

High precision mechanisms such as Scanning Tunneling Microscopes (STM) and Fast Tool Servos (FTS) use piezoelectric (PZT) actuators to achieve accurate and repeatable motion on the submicron level. While PZTs offer many advantages including high operating frequencies and high stiffness, they are also subject to significant hysteretic losses between the applied voltage and the output displacement. These nonlinearities in the output response can affect the performance of the actuator.

A control scheme is presented in which nonlinear effects are reduced using a *compensator* module in conjunction with a linear controller. The compensator predicts an instantaneous gain of the nonlinear system and compares the gain to an average linear gain. The compensator then uses the ratio of the linear gain to the actual gain to either boost or reduce the controller signal. Experimental results show a 50% reduction in the form error using this compensation scheme on a non-rotationally symmetric optical surface (100 μm amplitude tilted flat) cut with a FTS.

COMPENSATION TECHNIQUE

As seen in Figure 1, the relationship of the output displacement of a PZT to the input voltage is nonlinear. Whereas a linear actuator can be modeled as a fixed gain, the PZT represents a varying gain, the value of which is the slope of the hysteretic curve at the sample time, or K_{actual} . Since most controller theory is based on the assumption of linear plant dynamics, hysteresis makes it difficult to accurately control a PZT actuator.

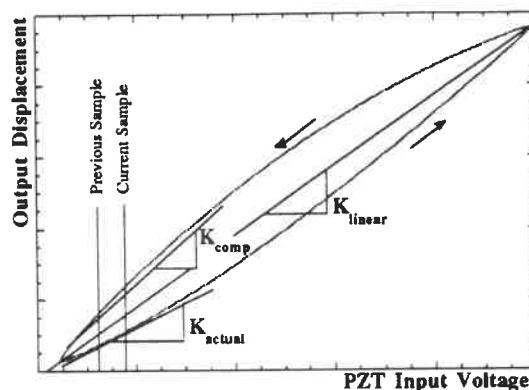


Figure 1: PZT Hysteretic Response

The technique proposed is shown in the block diagram of Figure 2 and is similar to a Time Delay Controller approach [1]. The relationship between the input voltage from the amplifier and the servo displacement is monitored in the

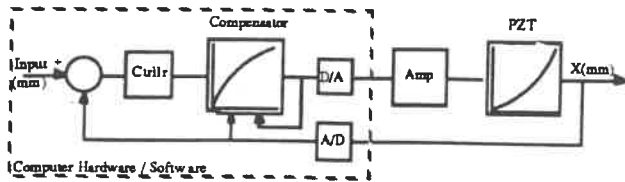


Figure 2: Compensation of the PZT Actuator.

compensator. The actual slope $\left(\frac{\Delta \text{ displacement}}{\Delta \text{ voltage}}\right)$ is compared to the linear slope of the piezoelectric and the ratio is used to calculate a compensating slope, K_{comp} , which solves the relationship

$$K_{comp} * K_{actual} = K_{linear}.$$

By multiplying the controller signal by this compensating slope, the nonlinear gain of the PZT (K_{actual}) is corrected and the linear gain, K_{linear} , should be achieved.

Inversion of the hysteretic effect has been previously implemented [2], but required *a priori* knowledge of the system. A major advantage of the technique described here is that the compensator is able to react to changes in the system response depending on the speed of the controller. For instance, the PZT hysteresis changes with frequency and amplitude of operation, but since the compensator calculates the slope in real time no prior knowledge of the system is required. The compensator, however, does operate on a time delay response; that is, the output from the controller is modified based on the *previous* response of the system and lags the input by one sampling period. Sudden changes in the plant response (such as reversal of the actuator) therefore require high sampling frequencies.

PERFORMANCE BENCHMARK

The MAC100 FTS was developed by the PEC for Oak Ridge National Laboratories to

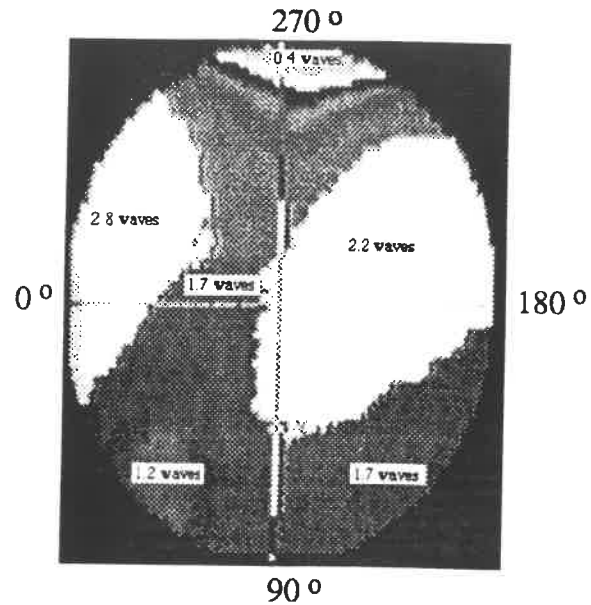


Figure 3. Tilted Flat without Compensation. Angles represent encoder readings.

fabricate non-rotationally symmetric optical components on a Diamond Turning Machine (DTM). It was designed for a maximum range of 100 μm at frequencies as high as 50 Hz.

A 100 μm tilted flat was cut in a 75 mm diameter copper workpiece at 1000 rpm (16.7 Hz) for benchmark testing. The tilted flat serves as a good benchmark because it requires full motion of the actuator and can be measured with a flat optical reference. The commanded displacement of the actuator is given by

$$z = \frac{A}{2} * \frac{r}{Rad} * \sin\phi$$

where A is the peak to valley amplitude of the tilted flat, r is the distance of the tool from the center of the workpiece, Rad is the workpiece radius, and ϕ is the angle of the DTM spindle.

Figure 3 is an interferogram of the tilted flat machined without compensation. The angles represent the spindle encoder angle when the tool is in that region of the workpiece. The

peak to valley form error is 2.85 waves, or 1.8 μm . Additionally, the arrangement of peaks and valleys is non-symmetric, with peaks at approximately 200° and 340°, and valleys at 60° and 270°. The profile across the center of the part in Figure 3 had 1 wave (0.6 μm) P-V form error horizontally and 2.2 waves (1.4 μm) vertically.

Figure 4 shows a plot of the hysteresis of the piezoelectric stack as measured from the output of the controller to the PZT displacement (refer to Figure 2). This perspective is used because the goal of the compensator is to linearize the system *from the output of the controller to the output displacement of the PZT*. A line drawn between the two endpoints of travel illustrates the nonsymmetry of the hysteresis. When the stack is extending, the hysteretic losses, represented by the deviation from the median line, are less than when the stack is relaxing. This may account for the lack of symmetry of the form error seen in Figure 3.

HYSTERESIS COMPENSATOR IMPLEMENTATION

The hysteresis compensator was implemented on the MAC100 as shown in Figure 2. Initially many problems were encountered with the compensator. The response of the PZT drifted sporadically and often clipped on reversal, and the displacement was generally noisy. Examination of the calculated values for K_{comp} , the compensation ratio, revealed noisy and discontinuous values. Since the K_{comp} value determines the amount of compensation, an accurate value is critical for the success of the compensator.

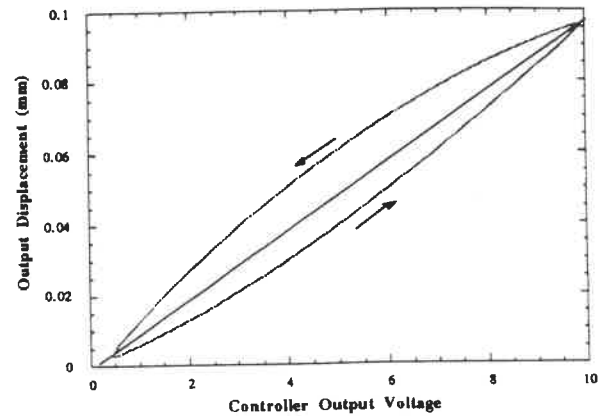


Figure 4. Uncompensated response of PZT.

Several reasons existed for the errors described above. Noise in the cap gage signal and the A/D and D/A converters has a direct influence on the value for K_{comp} . At the extremes of the input signal where the PZT reverses direction, $\Delta\text{voltage}$ and $\Delta\text{displacement}$ become very small and are susceptible to noise, causing K_{comp} to go to zero or infinity accordingly. To rectify this problem, a second order digital Bessel filter was implemented on both the voltage and displacement signals. Limits were also placed on the K_{comp} values. These changes had immediate affect on the compensated system response, making it more linear and smooth.

Figure 5 shows a plot of the remaining hysteresis after compensation and filtering. The width of the hysteresis is decreased significantly as compared to Figure 4, with most of the nonlinearity occurring at the end of travel. Part of the error at the reversal resulted from the compensator amplifying the controller signal past the maximum value of 10 volts, thereby clipping the signal.

Figure 6 shows an interferogram of the part cut on the MAC100 with compensation. The peak to valley form error has been reduced to 1.4

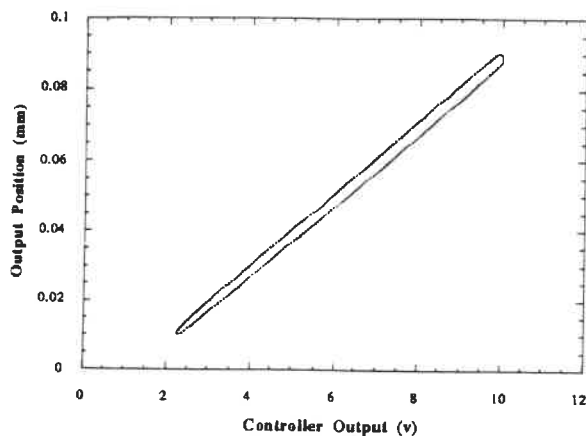


Figure 5. Compensated Response of PZT

waves, or $0.89 \mu\text{m}$. Also, the error has taken on more of a symmetric appearance, such that the part is "saddle-shaped" with low points at 90° and 270° . The profiles across the center have P-V form errors of less than 0.2 waves, ($0.13 \mu\text{m}$) horizontally and 1.3 waves ($0.82 \mu\text{m}$) vertically. This represents a 50% reduction in the form error as compared to Figure 3.

The remaining form error is the result of two phenomena. The first causes the saddle shape on the part and is most likely due to the saturation of the PZT at the extremities of its travel. The second is a disturbance that crosses the part diagonally from 80° to 260° . Since the part turns counter-clockwise, these regions of the part represent the areas just following reversal of the PZT. These errors were influenced by the controller gains and are probably due to a sharp disturbance in the control signal when the compensator gain changes suddenly on reversal. If the PZT remains extended too long at 90° , one would expect to see a "valley" at $\sim 80^\circ$, and remaining retracted too long at 270° should result in a "ridge" at $\sim 260^\circ$, both of which are seen in Figure 6. Tuning of the controller reduced this phenomena and minimized the error.

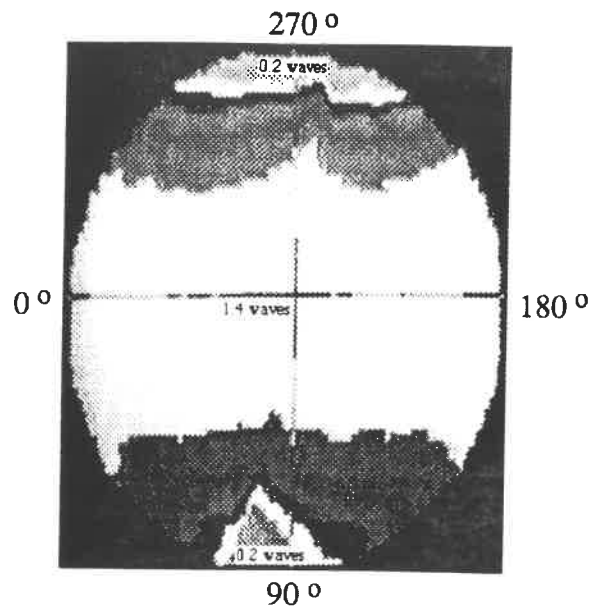


Figure 6. Tilted Flat with Compensation.

CONCLUSION

Compensation of hysteresis in piezoelectric actuators is important for achieving accurate linear motion. A scheme has been developed that compensates for these nonlinearities without prior knowledge of their form. Experimental results show a 50 % reduction in figure error in a $100 \mu\text{m}$ tilted flat optical surface when this technique is used.

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Analysis of Deformation of Air Spindle due to Centrifugal Force

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Summary;

This paper deals with numerical and experimental investigations on deformation of precision air spindle in diametrical expansion and axial contraction due to rotation. The deformation of spindle rotor is measured with special experimental setup and the results are compared with the FEM analysis. Special precautions are taken in the experiments to realize precision measurement in the order of sub-micro-meters, such as compensation of the thermal deformation of spindle and protection of the sensor probes against exhaust air flow. The experiments are carried out in a speed range up to 7000 rpm, and it is proved that the measured results agree well with the calculation.

1. Introduction

The air spindle is widely adopted in ultraprecision machining for its high rotational accuracy^{1),2)}, high rotational speed, low heat generation etc.. Since the accuracy requirements for the air spindle are quite high, its rotational error and deformations due to the centrifugal force, thermal effect etc. must be carefully examined even though their amount is small. The deformation of air spindle due to the centrifugal force during rotation is analyzed, and the experimental methodologies are developed here.

in the bearings during rotation is almost the same with that at spindle rest³⁾. The physical properties employed for calculation are summarized in Table 1.

An example of calculated deformation of spindle at rotational speed of 7000 rpm is shown in Fig.2. It is understood that both the diametrical

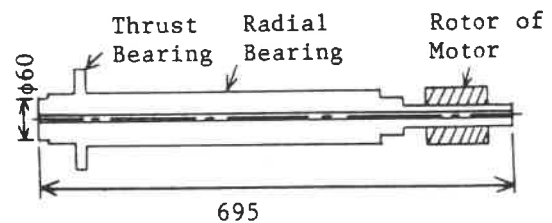


Fig.1 Spindle rotor

2. FEM Analysis of Deformation of Spindle Rotor

Figure 1 shows the schematic illustration of spindle rotor which is adopted for the present analysis. The axisymmetrical body is divided into 299 ring-type elements with rectangular sections and 369 nodes for FEM analysis. The axial displacement of the nodes at the middle of thickness of the thrust bearing is constrained to be zero in order to prohibit the movement of rigid body modes.

The deformation due to the change of air pressure distribution is neglected, since the pressure distribution

Table 1 Material properties

	Material	Density g/cm ³	Young's Modulus GPa	Poisson's Ratio
Spindle Rotor	Stainless Steel(JIS: SUS420J2)	7.75	205.9	0.3
Rotor of Motor(I) (Radius= 30-40mm)	Aluminum(45%)+ Magnetic Steel(55%)	5.45	152.0	0.31
Rotor of Motor(II)	Magnetic Steel	7.70	215.7	0.3
Taper Sleeve	Carbon Steel(JIS: S35C)	7.85	205.9	0.3

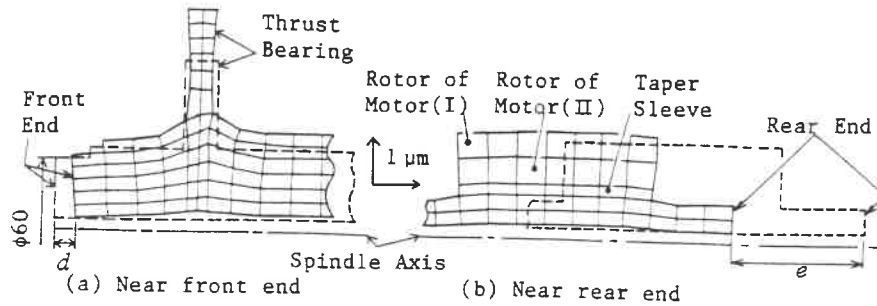


Fig.2 Deformation of spindle rotor calculated by FEM
 ----:At 0 rpm, —:At 7000 rpm

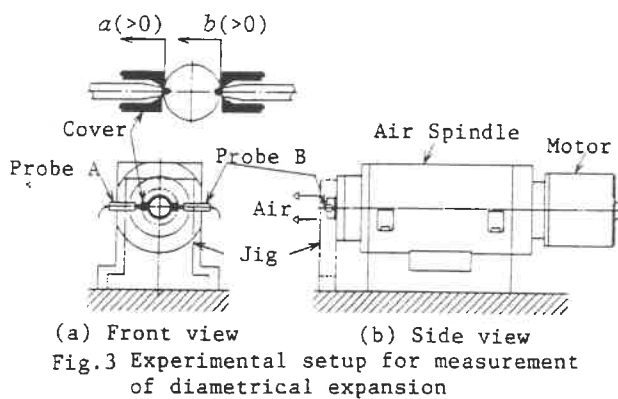


Fig.3 Experimental setup for measurement of diametrical expansion

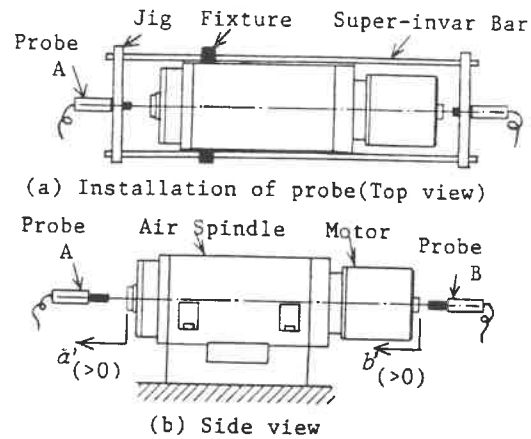


Fig.4 Experimental setup for measurement of axial displacement and expansion

expansion and the axial contraction take place at all nodal points of the spindle due to the centrifugal force. The diametrical expansion of the radial bearing is relatively larger near the thrust bearing due to a large diametrical expansion of the thrust bearing as compared to other portion of the radial bearing.

3. Experimental Analysis

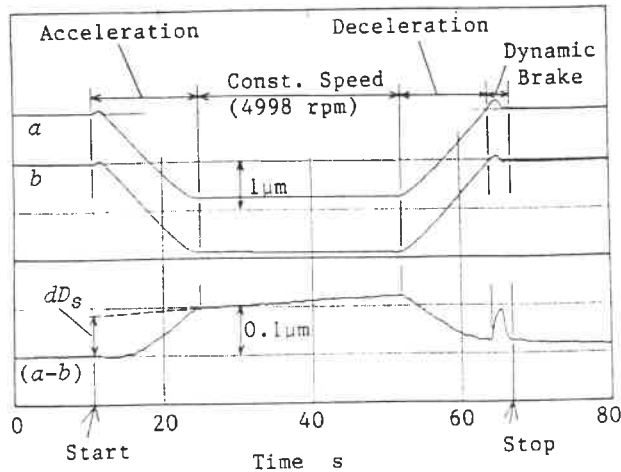
3.1 Experimental Method

Figure 3 shows the experimental setup to measure the diametrical expansion of the spindle rotor. A pair of capacitive-type proximeter probes are installed in horizontal direction as shown in Fig.3 at a location 5 mm from the front end of the spindle where the diameter of the spindle rotor is 60 mm. The sensor probes are covered with cylindrical insulators made of poly(methyl methacrylate) so that the

sensor probes are not directly exposed to the exhaust air. The output signals of the displacement meters are fed to the computer via an A/D converter after low-pass filtering with filters of cutoff frequency of 0.4 Hz.

In order to measure the axial deformation of the spindle rotor, the same pair of the capacitive-type proximometers are arranged at the front and the rear ends of the rotor as shown in Fig.4.

The rotational speed of the air spindle is limited to a maximum of 7200 rpm. The spindle was accelerated at a rate of 400 rpm/s after the start of rotation, and decelerated at the same rate of -400 rpm/s before the stop of rotation. The pressure of compressed air supplied to the spindle is 0.539 MPa.



(a) Example of recordings

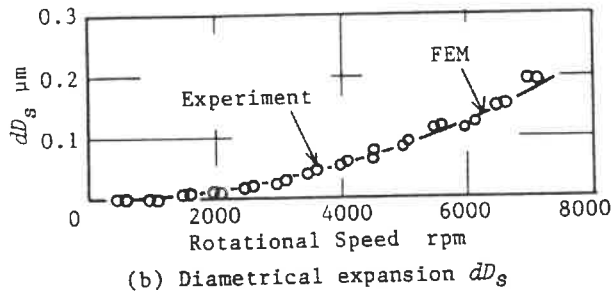
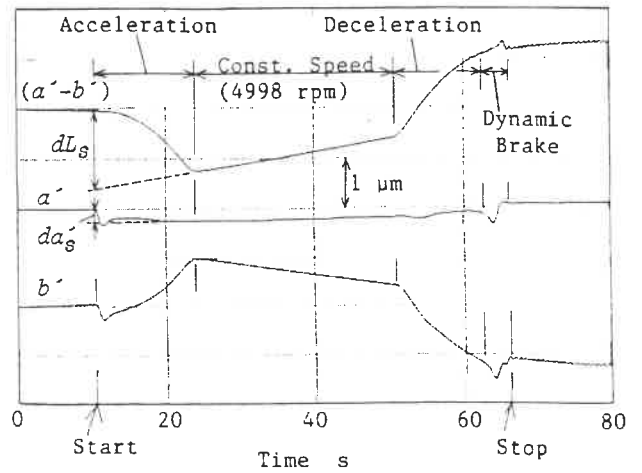


Fig.5 Diametrical expansion due to centrifugal force
(With low pass filter, with probe cover,
150 Hz sampling, measured diameter=60mm)



(a) Example of recordings of axial expansion

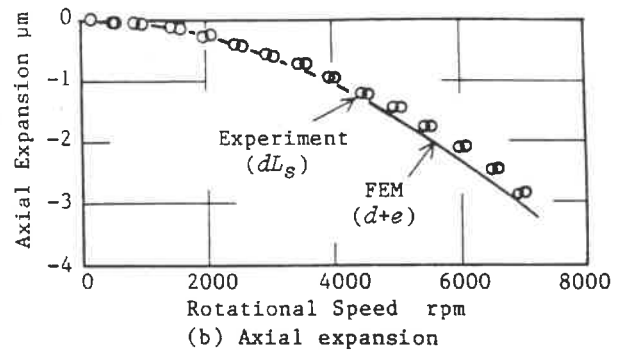


Fig.6 Axial expansion due to centrifugal force
(With low-pass filter, with probe cover,
150 Hz sampling, measured length=695mm)

3.2 Diametrical Expansion of Spindle Rotor

Examples of recorded displacements in the radial direction are shown in Fig.5(a). Displacements a and b measured with probes A and B respectively are expressed so that they have positive value when the spindle moves leftwards in Fig.3(a). It is understood from Fig.5(a) that the spindle rotor moves in the right direction as the spindle starts to rotate. The output $(a-b)$, which corresponds to the diametrical expansion of the spindle rotor, increases with an increase in the rotational speed during the acceleration period. It then increases gradually and almost linearly with the elapse of time during the constant rotational speed period.

The linear expansion of spindle rotor during the period of constant

rotational speed is due to the thermal expansion of the spindle rotor. In order to compensate the effect of thermal expansion, the plot obtained at constant speed was extrapolated and the pure diametrical expansion due to the centrifugal force was obtained which is designated as dD_s shown in Fig.5(a).

Figure 5(b) compares the diametrical expansion obtained experimentally and that calculated by FEM. It is understood that both the experimental and the calculated results agree well.

3.3 Axial Contraction of Spindle Rotor

Examples of recorded displacements in the axial direction are shown in Fig.6(a), where a' and b' represent the displacements at the front and rear of the spindle rotor respective-

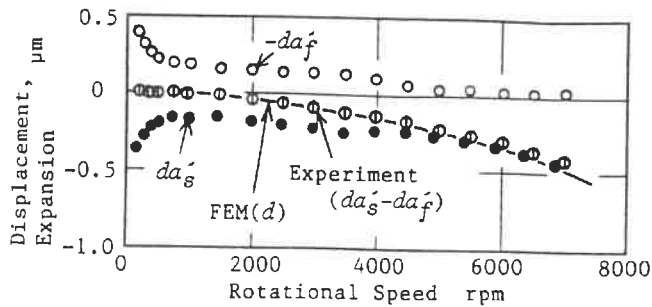


Fig.7 Displacement of front end of spindle and axial expansion d

ly. The same procedure was taken to compensate the thermal expansion of the rotor and the pure expansion of the rotor is obtained which is designated as dL_s shown in Fig.6(a).

In Fig.6(b), the axial expansion dL_s of the rotor measured at various rotational speeds is compared with the calculated values $(d+e)$ defined in Fig.2. It is understood that the measured expansion agrees well with the calculated one.

Displacement of the front end of spindle associated with the spindle rotation is designated da_s as shown in Fig.6(a). Displacement of thrust bearing da_f due to the driving force of motor is obtained as the difference in displacement when the spindle starts to run freely, or the driving force changes to zero. Thus the value $(da_s - da_f)$ gives the axial expansion of the portion between the thrust bearing and the front end of the spindle rotor, and it corresponds to d obtained by FEM shown in Fig.2(a). The results are summarized in Fig.7, and the experimental results agree well with the FEM analysis.

3.4 Precautions for Experimental Error

In order to improve the measuring accuracy in the experiments, special precautions are taken, which are; (1) compensation for the thermal expansion of spindle rotor, (2) low-pass filtering of measured signal, and (3) insulation of sensor probes against the exhaust air flow. Some of the details of such effects are discussed here.

As shown in Fig.5(a) the output signal $(a-b)$ increases almost linearly during the constant speed period. The linear expansion depends on the thermal expansion of the spindle rotor caused by the heat generated at the thrust and radial bearings and the built-in motor. In order to compensate the effect of thermal expansion, the diametrical expansion due to the centrifugal force is obtained by extrapolating the experimental results as was discussed in 3.1. The same compensation procedure is taken for the signal $(a'-b')$ in Fig.6(a).

It was experimentally found that the sensor signal is much influenced by the exhaust air flow when it is exposed to the air flow, and hence the sensor probes were covered with insulators as shown in Fig.3.

4. Conclusions

The deformations of air spindle rotor in the diametrical expansion and the axial contraction due to the centrifugal force during the rotation are analyzed by FEM and experiments. The following remarks are concluded.

- (1) The deformation of spindle obtained by experiments agree well with the FEM analysis.
- (2) Methodologies are established for precision measurement of spindle deformation, taking into account of the effects of thermal deformation, noise due to the air exhaust etc..

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Measurement of Mechanical Properties of Tympanic Membrane Using Force Sensing Operation Knife and Micro Testing System

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1. INTRODUCTION

While biomembrane has been pathologically or histologically studied, its mechanical properties have been paid little attention. During medical examination or surgical operation, doctors feel mechanical properties of biomembrane and are empirically aware of the relation between the mechanical properties and disease. Especially in otolaryngology, mechanical properties of tympanic membrane(TM in abbreviation) are desired, since its abnormal mechanical property could be a cause of serious disease. Therefore, measurement of mechanical properties of biomembrane is indispensable for future medical examination.

The authors newly developed two systems for measurement of mechanical properties of biomembrane, measured both of rupture stress and strain of TM, and evaluated the systems.

2. FUNCTIONAL REQUIREMENTS FOR MEASUREMENT

There are two kinds of measurement to grasp mechanical properties of biomembrane: One is measurement on living biomembrane. The other is accurate measurement on cut-off biomembrane.

Ideally, the measurement should be executed on living biomembrane. This is because mechanical properties deteriorate rapidly following death. The problem is that damaging living biomembrane only for measurement is generally unacceptable for ethical reason. Therefore, a measurement during medical examination or surgical operation is the most adequate. This measurement is defined "Clinical Measurement". Functional requirements for the measurement are as follows: (i) Measurement and operation should be performed simultaneously. (ii) Operation procedure, tool or specimen should be the same as usual operations.

Since the Clinical Measurement is performed by human operator, irregular operating condition makes quantitative evaluation difficult. To solve this problem, another measurement must be made under a certain condition and compensate the results of the above measurement. This measurement is defined "Laboratory Measurement". Functional requirements for the measurement are as follows: (i) Measurement conditions, such as membrane holding condition or rupture speed, should be accurately set. (ii) Very small force and displacement should be detectable, since the biomembrane is very thin ($17\mu\text{m}$ for guinea pig TM and $60\mu\text{m}$ for human) and weak. (iii) Real-time observation image of rupture process and measured data should be obtained at the same place and the same time.

3. CONFIGURATION OF MEASUREMENT SYSTEMS

Taking account of these requirements, the authors developed two systems named "Force Sensing Operation Knife" for the Clinical Measurement and "Micro Testing System" for the Laboratory Measurement to grasp mechanical properties of TM.

3-1 CLINICAL MEASUREMENT

Force Sensing Operation Knife, shown in Photo.1, was developed to satisfy the functional requirements for the Clinical Measurement. It detects piercing force of TM during piercing operation for otitis media treatment. Operation procedure mentioned below is the same as

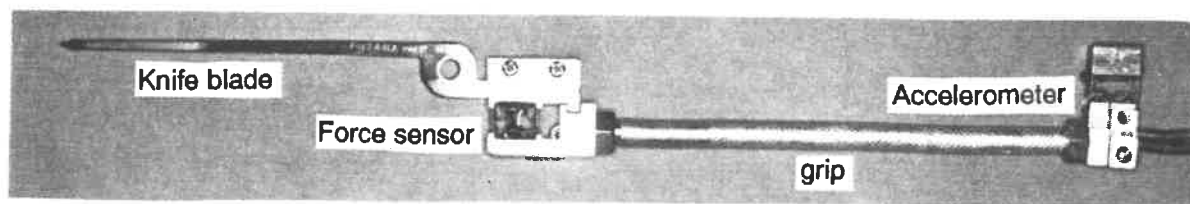


Photo.1 Force Sensing Operation Knife

usual operation: (i) A patient lies on bed with his ear-hole up. (ii) A doctor observes TM with microscope and piercing it only with thrust motion. It is almost the same size, weight and stiffness as usual operation knife and the doctor does not feel any difference. Usual disposable knife blade is attachable on it.

One-directional force sensor and one-directional accelerometer are installed between blade and grip and at the end of the grip, respectively. For both sensors, Parallel Plate Structure that the authors newly invented is used. Four strain gages are glued on the roots of two parallel thin plates. The resolution of the force sensor is 1mN . The role of accelerometer is to cancel the influence caused by hand vibration.

3-2 LABORATORY MEASUREMENT

Micro Testing System, shown in Fig.1, was developed to satisfy the functional requirements for the Laboratory Measurement. Measurement procedure is as follows (See Fig.2): (i) A TM is cut off and clamped by specimen holder. (ii) A round punch penetrates at the center of the clamped TM until it ruptures. (iii) Observing penetration process, force and displacement are measured.

The specimen holder clamps a cut-off TM between base and weight. Each has a hole of 0.5mm in diameter at the center. It spreads TM as the shape of a drum of 0.5mm in diameter.

The punch has cylindrical figure of 0.13mm in diameter and its tip is flat. Since its diameter is about the width of ten fibers of TM, influence of fiber density variance is reduced.

Penetration force acting on the punch is detected by a newly designed force sensor of $10\mu\text{N}$ in resolution. Its structure is the same as that of the Force Sensing Operation Knife. Displacement is detected by a linear scale of $0.1\mu\text{m}$ in resolution. Penetration speed is variable and its maximum is $1000\mu\text{m/s}$. Standard speed is $4.8\mu\text{m/s}$.

Stereo optical microscope is used for observation of rupture process. Its maximum magnification is 384. The microscope image through CCD camera appears on TV monitor and is recorded with the real-time displacement-force graph superimposed.

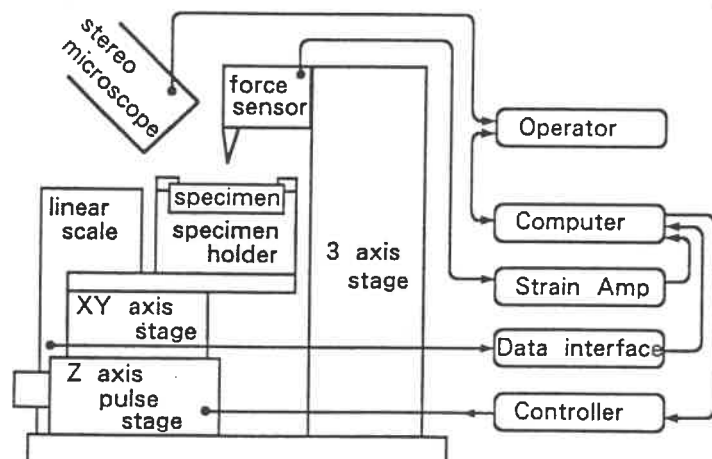


FIG.1 Configuration of Micro Testing System

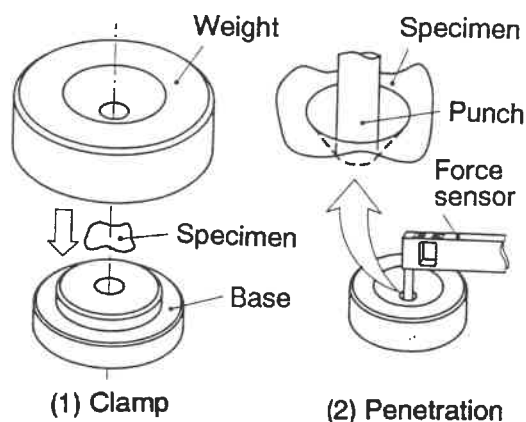


FIG.2 Procedure of Laboratory Measurement

4. EVALUATION OF MEASUREMENT SYSTEMS

4-1 CLINICAL MEASUREMENT

The Force Sensing Operation Knife is actually used in daily piercing operations of TM. The main result is mentioned below.

An example graph of piercing force during operation is shown in Fig.3. Following three phases appear here: (A) The knife blade touches the TM. (B) The knife edge is cutting the TM fibers. (C) The knife is drawing back. Piercing force is defined as its maximum point. Distribution of the piercing force is shown in Fig.4. The force scattered from 11 to 443mN. The data of TMs appearing normal, black portion in Fig.4, scattered little. TMs with smaller or bigger piercing force sometimes had serious disease. The results suggest the relation between piercing force and disease.

Piercing operation is sometimes performed to the same patient again. In this case, piercing force became bigger than that of former piercing. This result revealed that piercing operation might influence the mechanical properties of TM.

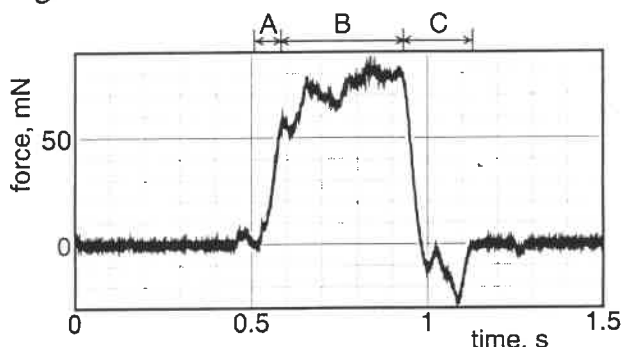


FIG.3 Example of piercing force during operation

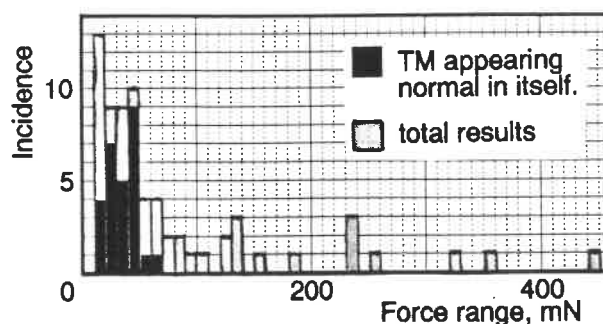


FIG.4 Distribution of piercing force measured by Force Sensing Operation Knife

4-2 LABORATORY MEASUREMENT

Using the Micro Testing System, sixty guinea pigs TMs and fourteen human TMs were measured. The main result is mentioned below.

The function of simultaneous display of the microscope image and measured data revealed rupture process as follows. (1) Guinea pig TM: Typical rupture process of guinea pig TM is shown in Photo.2. Their typical displacement-force diagram is shown in Fig.5. Rupture process was as follows: (i) Circular fibers ruptured first and only radial fibers sustained force in a shape of bridge as is shown in Photo.2. Some sudden dips in Fig.5 represent ruptures of circular fibers. Smaller quivering wave can be induced by ruptures of matrices. (ii) Eventually, radial fibers ruptured and the punch perfectly penetrated the membrane. (2) Human TM: Ruptured shape of a human TM is shown in Photo.3. Its displacement-force diagram is shown in Fig.6. Rupture process of human TM was different from guinea pig TM. After deformed to a cone shape, it ruptures like an open door. The diagram has no dip or quivering wave. These results suggest that its circular

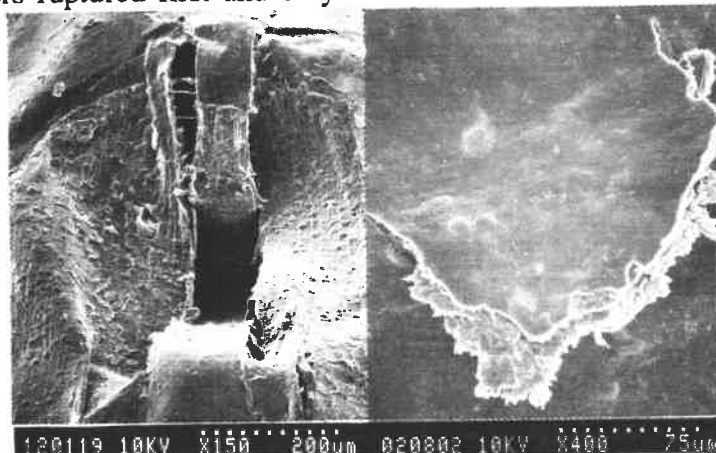


Photo.2 Rupture process of guinea pig TM

Photo.3 Ruptured shape of human TM

and radial fibers have almost the same toughness.

A series of Laboratory Measurements gave the following information about mechanical properties of TM. Guinea pig has 18MPa as rupture stress and 0.61 as rupture strain; human TM does 28MPa and 0.52.

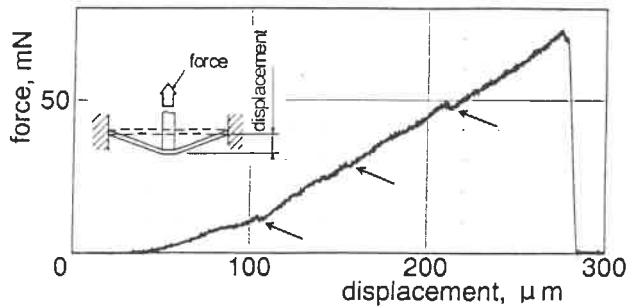


FIG.5 Displacement-force diagram of guinea pig TM measured by Micro Testing System

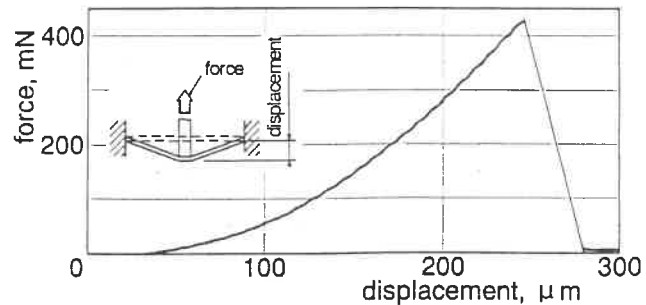


FIG.6 Displacement-force diagram of human TM measured by Micro Testing System

5. DISCUSSIONS

The results of the Laboratory Measurement do not coincide with that of the Clinical Measurement. Regarding the main cause of this discrepancy as the influences of initial tension and strain rate, the authors performed following two experiments.

(1) Experiment to clarify the influence of initial tension: Discrepancy of the mechanical data between TM clamped by the specimen holder and living TM was investigated. For this purpose, TM attached to its surrounding bone and auditory ossicles was tested by the Micro Testing System. Its rupture stress was less than half and rupture strain was about 1/6 of that of TM clamped by the holder. The results suggest that initial tension of holder-clamped TM must be smaller than living TM. Consequently, a specimen holder that can clamp TM with bone is to be developed.

(2) Experiment to estimate the influence of strain rate: Generally speaking, mechanical properties of viscous/plastic materials such as biomembrane show time dependence. Usually, this phenomenon appears as stress dependence on strain rate, stress relaxation, and age hardening. Using the Micro Testing System, stress relaxation of human TM was tested. The procedure is as follows: (i) Penetration is stopped before initial rupture. (ii) Holding the punch at the stopped position, the force measurement continues. The result is shown in Fig.7. Since it appeared much stress relaxation, to clarify stress dependence of TM on strain rate is inevitable. Consequently, the Micro Testing System with fast penetration speed as that of the piercing operation or physical impact in accidents is to be developed.

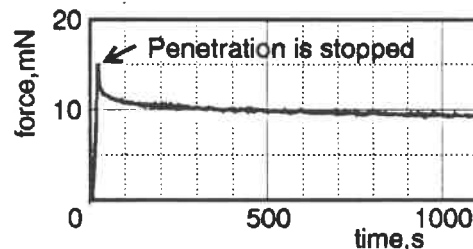


FIG.7 Stress relaxation of human TM

6. CONCLUSIONS

The Force Sensing Operation Knife and the Micro Testing System were developed in order to clarify mechanical properties of tympanic membrane. Through the several experiments, the authors confirmed the effectiveness of the systems concept.

ACKNOWLEDGMENT

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ON REPLICATION AND 3D STYLUS PROFILOMETRY TECHNIQUES FOR MEASUREMENT OF PLATEAU-HONED CYLINDER LINER SURFACES.

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SUMMARY

This paper concerns the use of replication techniques combined with 3D stylus profilometry for the monitoring of plateau-honed cylinder liner surfaces in engine manufacturing and testing. The replication technique has proved to be a useful tool for nondestructive testing of cylinder liner roughness and topography. The measurements were part of an extensive test program where changes in the manufacturing process had to be followed up by engine tests before full scale production was allowed. In those engine tests, surface roughness and topography were important factors to study. In the present investigation, 3D profilometry using a conventional diamond stylus was employed to map not only the profile section but also the liner's surface features in the plane (honing angles, smearing of coldworked material on the surface and wear development).

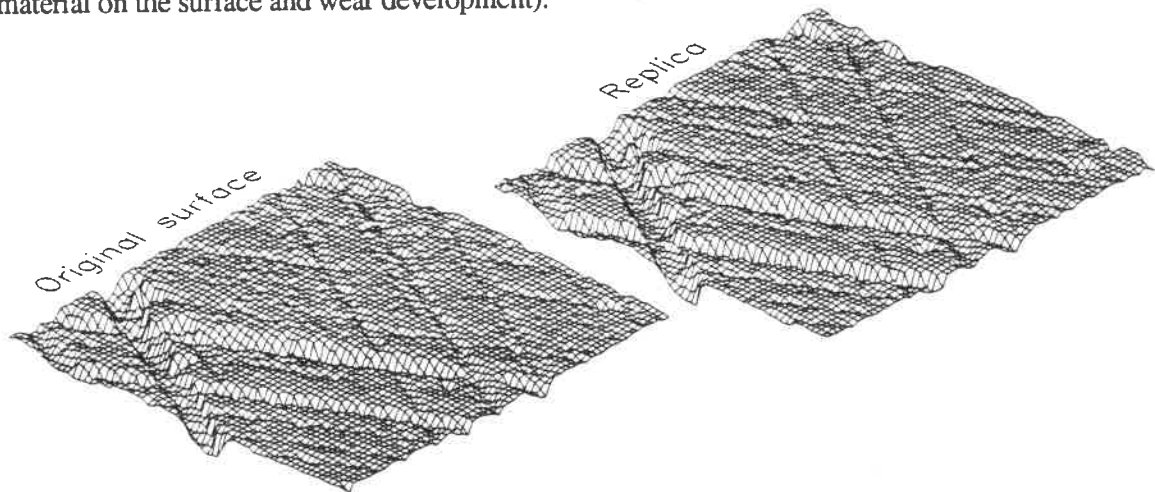


FIG. 1 Qualitative comparison of 3D roughness images of the original cylinder liner surface and its replica. By the use of 3D measuring technique, the corresponding areas are easily measured and compared by the identification of characteristic features in the surface plane e.g. scratches and pores (AutoCAD¹, B-G Rosén -93).

INTRODUCTION

One of the main concerns when working with replicas has been the fidelity of the replication technique itself. In order to investigate the influence introduced by the replication process, corresponding surface roughness measurements have to be carried out. The well-known problem with relocation and repositioning of the diamond stylus to exactly the same profile section both on the replica and on the corresponding original surface for the validation and comparison of 2D profile measurements [1] is

¹ Registered trademark of the Autodesk, Inc.

significantly reduced by the use of 3D mapping. Replication has been used in surface roughness measurements since the '40s [2] and a variety of different molding methods and materials has been investigated [1],[2],[3], throughout the years. The technique allowed engineers to measure surfaces where an inconvenient location (e.g. an engine assembled into a testrig), size (e.g. a ship's hull [4]), and E-modulus (e.g. soft materials) of an object prevented traditional stylus profilometric measurements.

The measurements presented in this abstract show good agreement between roughness parameters and visual appearance (fig. 1-2). Integrating (S_a) and extreme profile (S_z) parameters show differences around 5% for 3D roughness measurements on corresponding areas.

MATERIALS & METHODS

By letting a liquid two-component acrylate plastic² set and harden on the surface, a negative replica of the original surface is made. A lot of objections can be made about the accuracy of the replica and wavelength range of fidelity, due to molding material (grain size and shrinkage) and process (heat generation when hardening and surface reactions between replica material and the replicated surface) [3], [5]. However, in this paper we present an areal comparison method for the plateau-honed cylinder liner and the replica of the same area. The method employed to validate the replication process has been both a qualitative comparison of raster-scanned images [6] (fig. 1) and a quantitative comparison of 3D roughness parameters characterizing both extreme (S_z) and averaging parameters (S_a , S_q) as well as parameters describing the bearing area curve (S_k , S_{vk} , S_{pk}) suggested by Stout et al. [7]³ (fig. 2).

This areal comparison between the original cylinder liner surface and its replica was possible due to the 3D surface roughness measuring technique, where 3D mapping of the two made it possible to recognize the same characteristic features or landmarks (e.g. a pattern of scratches or pores) in both surfaces.

A study also was carried out on a engine subjected to a motor test in order to verify changes in the machining process. Here, a nondestructive marking of the surface had to be done and by the use of a 4-step process we managed to relocate to the same surface spot of the cylinder liner after the engine test, enabling us to retrieve measuring data on the cylinder wear progress.

- *Step 1:* Location of the clay mold in the liner by a simple fixture (here, we used a slice of rubber from an eraser aligned with the cylinder liner top, which served as a vertical reference) whereupon molding is done.
- *Step 2:* Rough detection and marking of areas for measurement by the identification of characteristic scratches visible in the replicas before and after the test by the use of a magnifying glass.
- *Step 3:* Exact identification of corresponding areas for comparison by 3D roughness measurement and contour map analysis.
- *Step 4:* Evaluation of parameters and depiction of the areas selected in step 3.

² In this study we used the Technovit 3040 by Kulzer & Co GmbH in Germany.

³ The parameters denoted with an S are 3D surface roughness parameters corresponding to the DIN 4776 and ISO standardized 2D profile parameters R_z , R_a , R_q , R_k , R_{vk} , R_{pk} .

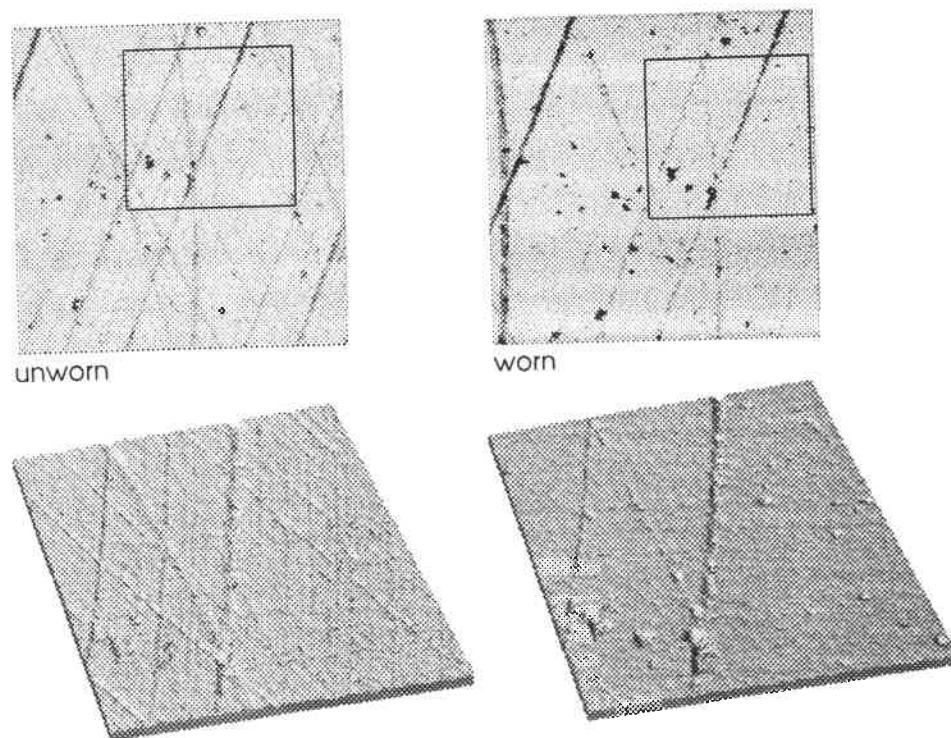


FIG. 2 Contour maps and 3D views of replicas from a worn and an unworn cylinder liner surface illustrating the relocation technique for comparison of surfaces in three dimensions (Rastroscope⁴, B-G Rosén -93).

RESULTS

The cylinder liner surface consists mainly of two superimposed components: the fine upper plateau component and the coarser valley component. The replication technique employed in this study shows a maximal increase of 15% for the valley component (S_{vk}) parameter value, while plateau parameters (S_k) present a deviation of 20% (fig.2). The industrially frequently used average parameters (S_a and S_q) as well as the peak to valley parameter (S_z -DIN) are even better replicated (5-10%).

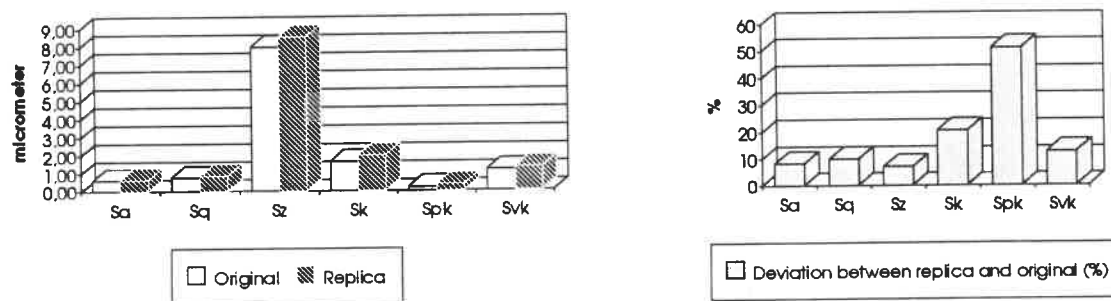


FIG. 3 A quantitative comparison of 3D roughness parameters [7] between the original plateau-honed cylinder liner surface (100 x 100 points) and the replica of the same 1mm² of surface.

⁴ Registered trademark of Danish Micro Engineering A/S.

The parameters calculated from the replica are consistently higher than the original measured parameters which have been established elsewhere [5], [8]. Much of this deviation resides in the fact that a stylus responds differently to a valley bottom than to a peak. The finite radius of a stylus prevents it from reaching the actual bottom of a narrow valley whereas a peak is easier to measure. It is therefore possible to detect fine valleys, invisible to the stylus in the original surface, since such valleys are transferred into measurable peaks by the replication technique.

DISCUSSION

The use of replication technique in this study has proved to be very useful from many points of view. With this technique, on-line measurement time in the test rigs is reduced and replicas of the tested surfaces can be stored and re-used for studying the different phases of the irreversible wear process of the cylinder liner in order to verify the changes in the honing process. Also, the use of 3D roughness measurements will be useful to locate corresponding areas for comparison and development of the replication method and replication materials. The results presented show an acceptable variation between original and replica surfaces and further studies might prove that valley components of a surface more accurately can be measured by a stylus on replicated surfaces than on original surfaces.

ACKNOWLEDGEMENT

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EFFECT OF TOOL WEAR ON SURFACE FINISH, TEXTURE, AND TOOL FORCES

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INTRODUCTION

Diamond machining of a surface is known to involve a complex interaction among many contributing factors, some controllable, others less so. One of the latter is the cutting tool geometry. At the resolution required in the diamond turning of optical quality components, the changing cutting edge geometry as the tool wears during its service life would be expected to have a pronounced influence on the process. Tool wear has been shown to affect surface roughness and machining forces[1], and the use of these forces as predictors of finish and for production tracking of wear state has been proposed. Other studies have suggested that nonlinearities in the machining process will produce in the aperiodic, high frequency part of the surface finish an anisotropic texture[2,3]. Texture should also be affected by tool wear. Further, this texture exhibits a fractal nature, and the fractal dimension in various directions on the surface should be useful as a measure of the degree of anisotropy and as a general means of characterizing surfaces. By examining how each of these parameters is affected by tool wear, some insight into any underlying causal relationships or interdependence between the variables may be gained.

TEST CONDITIONS AND METHODS OF MEASUREMENT

Test Conditions

Discs of electroplated copper on brass, 6061-T6511 aluminum, and 6061-T6511 which has been annealed (Vickers hardnesses of 1.6, 1.2, and 0.4 GPa, respectively) are cut concurrently at a position 65 mm radially from the spindle center. Programmed machining conditions are:

Spindle Rotation Rate - 200 rpm
Feed - 5 μm

Depth of Cut - 10 μm
Coolant - Mobilmet Omicron

Cutting is stopped after data collection to leave a step on the surface for measurement of the actual depth of cut with a Talysurf 5-120 stylus profilometer.

A 3.18 mm nose radius, 0° rake, 6° clearance angle diamond tool is used in all tests. This tool is worn by cutting a disc of 6061-T6 aluminum a specific distance between each test. Wear states selected for study are 0, 1, 3, 7, 15, 50, 100, 150, and 200 kilometers cutting distance. Following each controlled wear cut, the tool edge radius and clearance wear land are measured using a scanning electron microscope and a contamination deposition technique[4]. Figure 1 shows a contamination line crossing the edge radius of the test tool after 1 kilometer cutting distance.

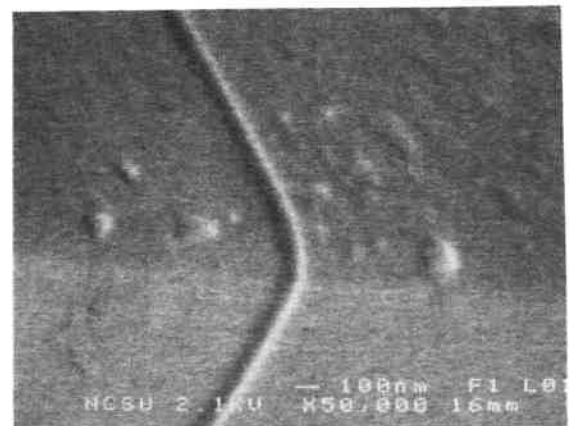


Figure 1: SEM micrograph of a line of contamination on the tool clearance face (top), edge radius, and rake face (bottom) after 1 km cutting distance.

Force Measurement

The cutting and thrust forces applied to the tool are measured using a Kistler 9251A piezoelectric transducer and two Kistler 5004 amplifiers. Amplifier output is sampled at 10 kHz with a computer, which scales and displays the force signals and records the data for later analysis. Several data files of 2 seconds duration are collected when the tool is cutting at the sample centers. Figure 2 reproduces a portion of a force signal. The signal baseline corresponds to the time the tool is not in contact with a sample. The initial 1 millisecond of signal after the tool contacts the workpiece is highly oscillatory and is not used in determining the forces.

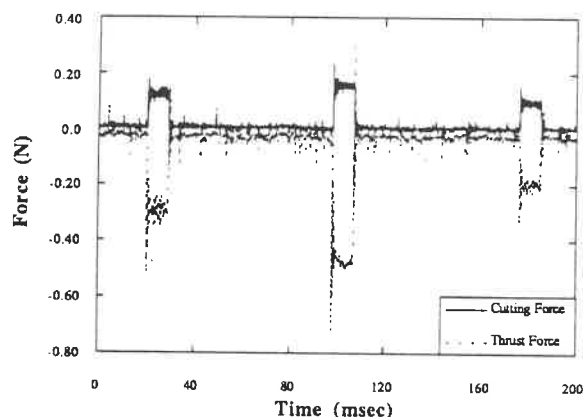


Figure 2: Cutting and thrust force signals for the heat-treated aluminum, copper, and annealed aluminum, respectively, at 1 km tool wear.

Surface Finish Measurement

Surface finish is measured with a Zygo Maxim 3D interferometric microscope using a 10X Mirau objective. Images are acquired at six locations following the path of the tool through the center of each sample. The area of tool entrance onto the surface, where tool vibration gives oscillatory forces, is avoided during image acquisition. Each image provides identifying information, isometric and contour plots of the surface, and the calculated arithmetic average roughness (R_a), root-mean-square roughness (RMS), and peak-to-valley roughness (P-V).

Characterization of Surface Texture

The Zygo image files are imported into a computer program which calculates their power spectra and analyzes the fractal characteristics of each. The software does not consider the periodic portion of the power spectra, indicative of the tool feed marks, since this would mask the underlying chaotic nature of the surface[5]. Anisotropies in the surface texture appear as changes in fractal dimension as direction along the surface varies. However, instrumental errors in the interferometric microscope affect this analysis for samples of finer finish. The roughness of surfaces cut to date in this study have not yet increased to levels sufficient to allow neglect of the instrumental errors, and a satisfactory method of error removal has not yet been determined.

PRELIMINARY RESULTS

The test tool geometry has been measured and cutting tests performed at the 0,1,3, and 7 kilometer wear states. Figure 3 shows a nonlinear increase in tool edge radius and wear land length. The values presented here are consistent with those reported earlier[1] for another round nosed tool.

Surface roughness values from the cutting tests are presented in Figures 4, 5, and 6. Each value is an average of the six images taken on each sample. These data do not reflect the improvement in roughness often seen during

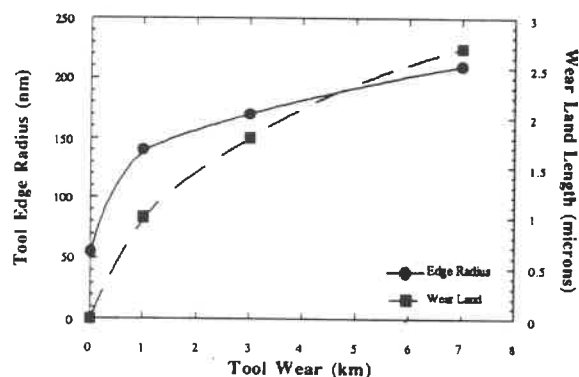


Figure 3: Increase in tool edge radius and wear land length as tool wear progresses.

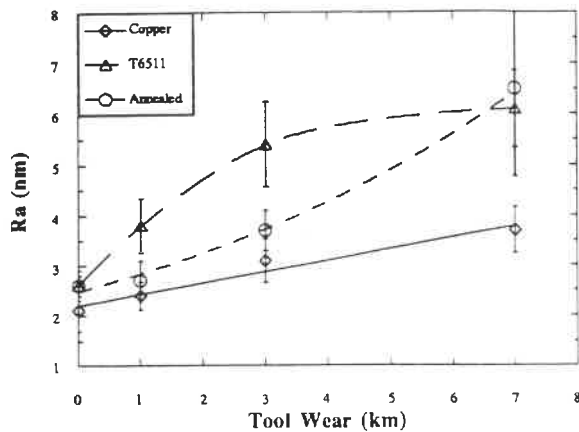


Figure 4: Average roughness as a function of tool wear.

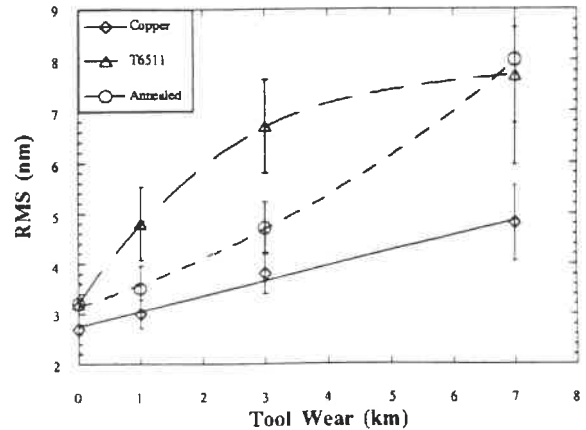


Figure 5: Root-mean-square roughness as a function of tool wear.

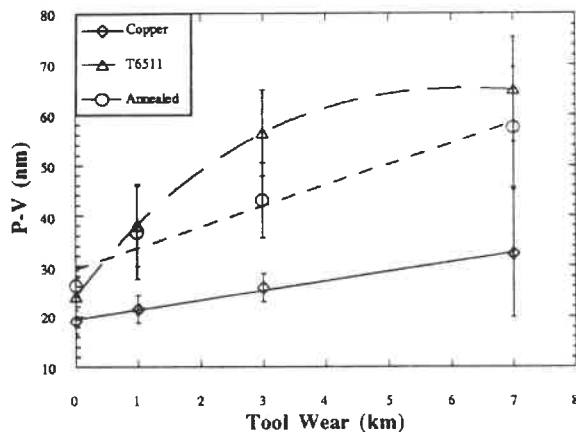


Figure 6: Peak-to-Valley roughness as a function of tool wear.

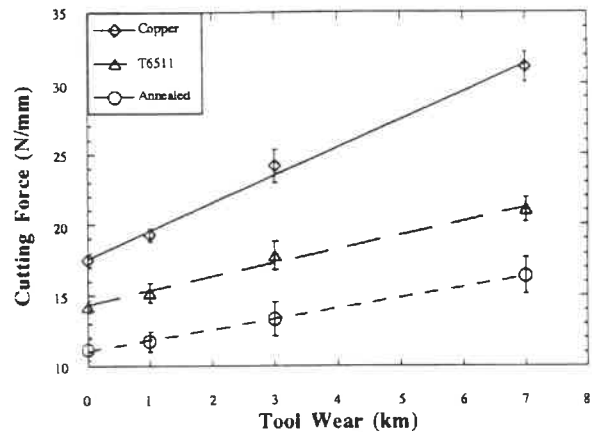


Figure 7: Normalized cutting forces as a function of tool wear.

early stages of tool wearing, though this effect was seen during earlier cutting tests with the test tool. This suggests factors other than tool wear may contribute to or hinder appearance of this phenomenon.

To allow a more direct comparison between materials and with other works, tool forces are normalized by dividing by the measured depth of cut. These normalized forces are shown in Figures 7 and 8. As expected, harder materials generate greater machining forces. The cutting forces are seen to increase moderately in a linear fashion during these early stages of wear with the slope or rate of increase steepening as material hardness increases. The thrust forces

increase appreciably and nonlinearly, though with the same relationship between hardness and rate of increase.

In Figure 9 the ratio of thrust force to cutting force can be seen to increase to a value greater than unity between the 0 and 1 kilometer wear states, corresponding to the period of formation and rapid growth of the wear land on the tool clearance face.

FUTURE EFFORTS

Cutting tests will be performed at the remaining cutting distances. Additional states may be added should examination of the data suggest

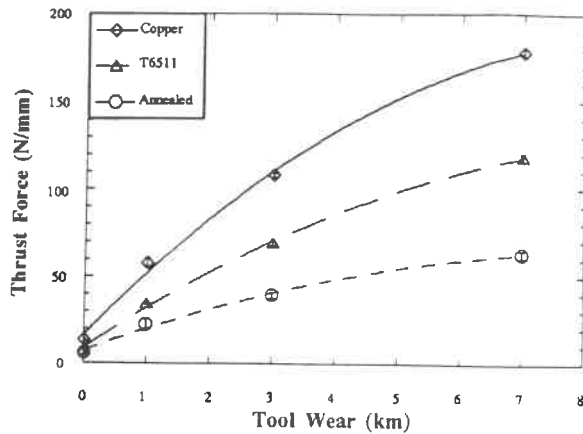


Figure 8: Normalized thrust forces as a function of tool wear.

this course. Efforts will be made to correlate the measured edge radius and wear land length with the observed forces and finish. Texture information via the power spectra will continue to be pursued and correlations with the finish, tool forces, and wear condition will be sought.

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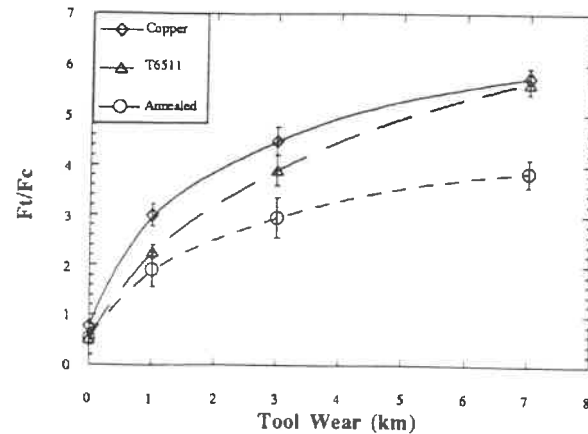


Figure 9: Ratios of thrust force to cutting force as a function of tool wear.

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DYNAMIC ERROR CORRECTION IN DIAMOND TURNING

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INTRODUCTION

An essential ingredient in the fabrication of diamond turned surfaces is the accurate control of the relative position of the tool and the workpiece. The existence of motion errors contributes to surface finishes being often an order of magnitude greater than the theoretical finish of an ideally controlled process. The control to this level (which to its credit has made diamond turning an extremely successful process in precision fabrication) has been achieved largely through careful machine design and production of accurate high stiffness slides and spindles. Performance has been extended beyond this basic level of machine accuracy by mapping repeatable "static" errors such as slide straightness and squareness, and compensating for them in the control computer. However, non-repeatable dynamic positioning errors such as slide vibration and spindle runout have for the most part remained at the level dictated by mechanical design and the disturbances present. These dynamic errors generally result in small amplitude (< 200 nm) variations in position at frequencies below 200 Hz. A high bandwidth (1000 Hz) piezoelectric fast tool servo stands out as being perfectly suited for correcting these small amplitude positioning errors. Such a capability was included in the design of the Large Optics Diamond Turning Machine at Lawrence Livermore National Lab [1]. In a more common setting of commercially available machines, the effectiveness of using a fast tool servo to correct for relative tool/workpiece errors on surface finish was demonstrated by Fawcett [2]. Though successful in improving surface finish, the technique employed was limited

to producing flat parts and did not account for the possibility of angular spindle runout.

A more general approach is presented which does not compromise the machine's versatility. The scheme utilizes laser interferometric feedback of slide position to provide measurement of slide vibration. Also, two eddy current probes observe the back surface of the spindle chuck to monitor workpiece motion errors due to axial and angular spindle runout (both synchronous and asynchronous). Using the sum of these two error measurements, the position of the tool is adjusted accordingly. With this scheme, it is desired to correct for the most significant sensitive direction dynamic positioning errors, allowing surfaces to more closely approach the theoretical surface finish.

DYNAMIC ERRORS AND THEIR EFFECTS

The sources of dynamic positioning errors on the diamond turning machine (DTM) at the Precision Engineering Center are summarized in Table 1. For most facing operations, the z-direction is the sensitive direction, so positioning errors in this direction are most significant because they directly alter the depth of cut.

Z-DIRECTION ERROR CORRECTION USING FTS

The most prominent z-direction error sources are the z-slide vibration and spindle runout. Runout errors affecting z-direction positioning include both axial and angular spindle motion. Angular spindle errors will be assumed to be important

DYNAMIC POSITIONING ERRORS

	SOURCE	MAGNITUDE
z-direction (sensitive direction)	z-slide vibration	25 nm - 100 nm (holding position) (higher feedrates)
	axial spindle runout	25 nm (sync.) 60 nm (async.)
	angular spindle runout	50 nm (@ max radius)
	x-slide and tool post vibration	<5 nm
x-direction (non-sensitive direction)	x-slide vibration	20 nm - 80 nm (holding position) (higher feedrates)
	z-slide vibration	<5 nm
	radial spindle runout	40 nm

Table 1: Dynamic positioning errors on the PEC's DTM

to the machining process only in the x-z plane (the plane of tool and workpiece interaction - see Figure 1). Angular motions in other directions will cause extremely small changes in rake angle which should be insignificant.

System Configuration

The layout of the DTM slide system incorporating the proposed correction scheme is shown in Figure 1. The z-slide position is measured by laser interferometer for slide position feedback, and is available for correction of slide vibration. The spindle runout is measured by mounting probes to the spindle bearing housing to monitor the position of the back surface of the chuck. In doing so, it is assumed that the spindle housing is rigidly fixed to the z-slide so that the motion detected is the motion of the spindle relative to the z-slide. The sum of this motion and the vibration detected by the laser feedback is then the motion of the spindle with respect to the base of the machine. If vibration of the tool post and x-slide in the z-direction are neglected, then this can be considered the motion of the spindle with respect to the tool (in the z-direction).

To be able to sense and correct for angular motion of the spindle, two probes will be used at tool level ($y = 0$), each the same distance

from the center of the spindle ($x = \pm L$). At any given instant in time, the error in the z-direction caused by axial and angular runout will be linear along the x-direction. The average of the two motions gives the axial runout and is the z-intercept of this line. The slope of the line is given by the difference in the motions divided by the distance between the probes. i.e.

$$\Delta z_{spin} = \frac{B - A}{2L}x + \frac{B + A}{2} \quad (1)$$

This would be a suitable measurement of the error if the back surface of the chuck (the target area for the probes) was perfectly flat. However, since this is not the case, it is necessary to map the errors in this surface as a function of the angular position of the chuck θ , so that the measured signal can be corrected. Modifying Eq. 1 to include the error mapping gives

$$e_{spin} = \frac{B - B_m(\theta) - [A - A_m(\theta)]}{2L}x + \frac{B - B_m(\theta) + A - A_m(\theta)}{2} \quad (2)$$

Eddy current probes are used for the displacement sensing. Since the signal produced by this type of sensor is subject to errors from variations in surface finish or metallurgical impurities

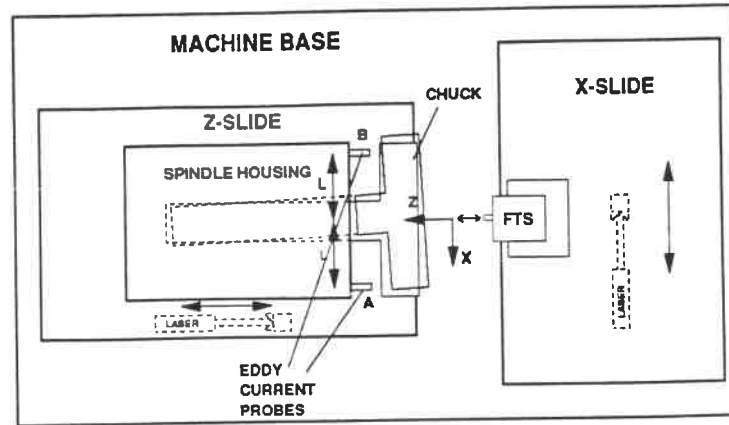


Figure 1: DTM slide system with z-direction dynamic error compensation

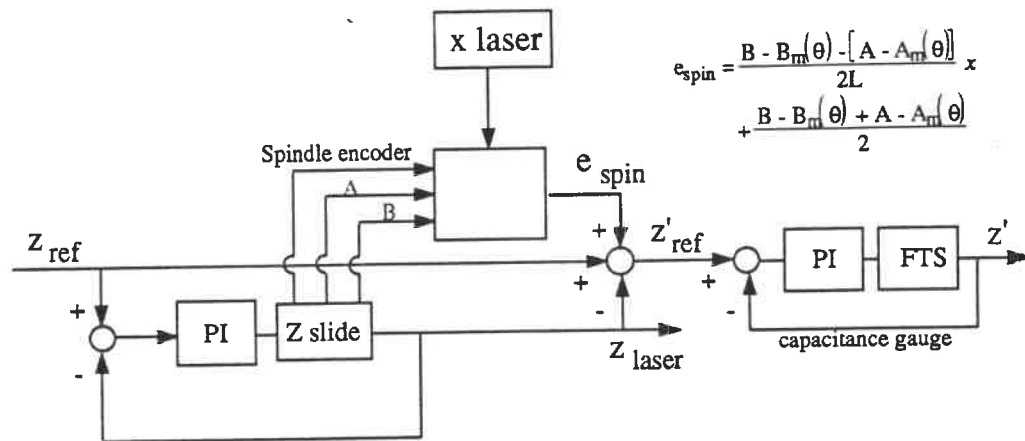


Figure 2: Schematic of z positioning control scheme

in a varying target area, the mappings $A_m(\theta)$ and $B_m(\theta)$ are mappings of the sum of the surface error and probe error for the given target area.

A schematic of the control scheme for the z-direction is shown in Figure 2. The error in slide position ($z_{ref} - z_{laser}$) is added to the error from the spindle to produce the reference position for the fast tool servo (z'_{ref}). The slide motion control loop operates with a 1 kHz sampling frequency, while the FTS loop operates at 10 kHz. In each 100 μs cycle, the eddy current probes, both lasers, the spindle encoder, and the FTS capacitance gauge must be sampled and the error

mappings must be read before the FTS control signal can be calculated.

Surface Error Mapping

The signal detected by the displacement sensors is the sum of essentially three effects. These are displacement due to axial spindle motion, displacement due to angular tilting of the surface, and the error in surface flatness (combined with material variation error for the case of eddy current probes). The last of the three of these should remain unchanged throughout virtually all operation of the machine. It is desired to sep-

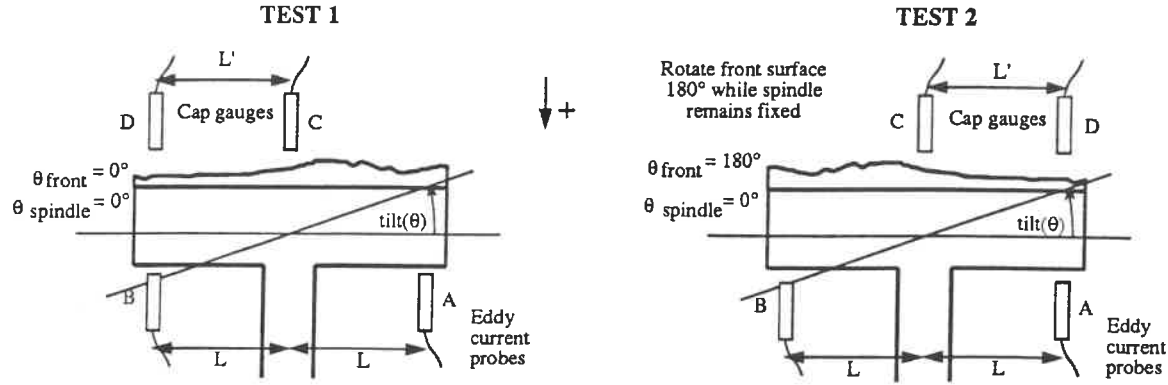


Figure 3: Spindle mapping setup

arate this constant component of the error from the others in an off-line measurement so that the other two varying components can be determined on-line during operation.

The setup for the mapping procedure is shown in Figure 3. In addition to the two eddy current probes (A and B) that will be used during operation, two capacitance gauges (C and D) are needed for the mapping. All four probes are mounted from the spindle housing. Throughout the mapping procedure, the axial spindle motion is measured directly using capacitance gauge C to sense the motion of the front face at the center of rotation. It is necessary to perform two separate "tests" in order to separate the tilting errors from the surface errors. Throughout each test encompassing one spindle revolution, measurements are to be taken of the back of the chuck and of a flat plate mounted to the front of the chuck. For the second test, the front plate is mounted in an orientation 180° from that of test #1. Also, capacitance gauge D is moved to the opposite side of center, so that it will be observing the same area during each test for any given spindle position.

A best fit sine wave (frequency of one spindle revolution) is subtracted from the data taken with capacitance gauge D to remove any tilt error introduced in mounting the front plate to the chuck. The test #2 data C_2 and D_2 are zeroed based upon the change in the initial values of A and B from test #1 to test #2. The mappings $A_m(\theta)$ and $B_m(\theta)$ are then calculated as follows.

$$\Delta tilt = \frac{B_2(\theta) - A_2(\theta)}{2L} - \frac{B_1(\theta) - A_1(\theta)}{2L}$$

$$tilt_1(\theta) = \frac{D_2(\theta) - D_1(\theta) + C_1(\theta) - C_2(\theta)}{2L'} - \frac{1}{2}\Delta tilt(\theta)$$

$$A_m(\theta) = C_1(\theta) + A_1(\theta) + L tilt_1(\theta)$$

$$B_m(\theta) = C_1(\theta) + B_1(\theta) - L tilt_1(\theta)$$

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Fabrication of the Attachment Rails Used for Mounting an Array of Eight X-ray Reflection Gratings

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Key Words: X-ray Multi-Mirror project (XMM), diamond-turning, snap-together optics, CBN machine, Large Optics Diamond Turning Machine (LODTM), error budget

Introduction

This paper describes the fabrication of a set of four attachment rails—parts that resemble precision step gauges. The attachment rails provide the precision mounting surfaces for a prototype array of eight X-ray reflection gratings for the European Space Agency's (ESA) X-ray Multi-Mirror project (XMM). Each rail is 4.5" long with a cross-section of less than 0.1 in², and has eight protruding bosses spaced approximately 0.5" apart (Figure 1). A diamond-turned feature on each boss provides a mounting surface for one of the four corners of a grating. These surfaces are 0.018" high by 0.1" wide, and have a 12" cylindrical radius with an axis parallel to the boss protrusion (Figure 2). Together, the four rails provide eight sets of four co-planar points for mounting the gratings (Figure 3). Note that the gratings are not parallel to each other; they sweep through a 12 mrad angle from the first to eighth grating. To accommodate this fanned array, the normal directions (denoted by arrows in Figure 1) of the mounting surfaces on the bosses, at the rail centerline, also sweep through a 12 mrad angle from the first to eighth boss.

The optical performance requirements of the grating array required that the mounting points locate the corners of the gratings within $\pm 15 \mu\text{m}$ of their designed positions. By virtue of the accurate location of the mounting surfaces with respect to each other, the rails allow the gratings to be arrayed with the spacing and orientation

accuracy necessary in a snap-together optics fashion; adjustments after installing the gratings are not required.

The base material of the rails is Ni-Fe Alloy 42 (invar), which has the same coefficient of thermal expansion as the gratings and the support structure ($2.4 \mu\text{m}/\text{in}\cdot^\circ\text{F}$). The bosses were plated with 0.005" of copper to allow single-point diamond-turning of the grating mounting surfaces with minimal tool wear.

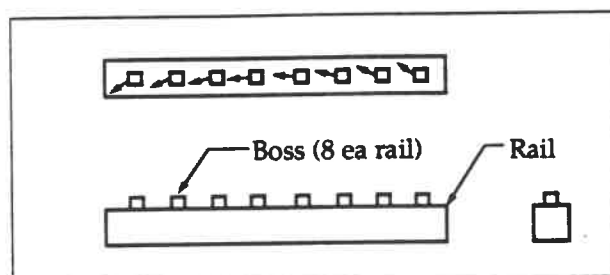


Figure 1. Rail with eight bosses.

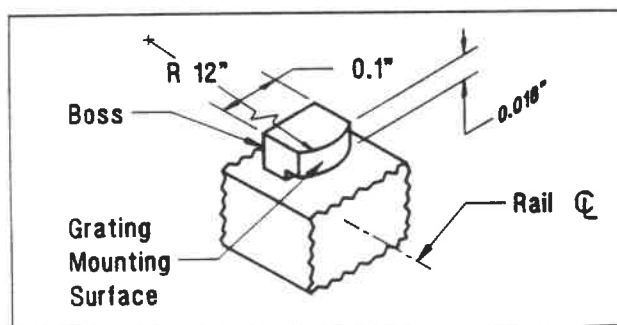


Figure 2. Boss detail.

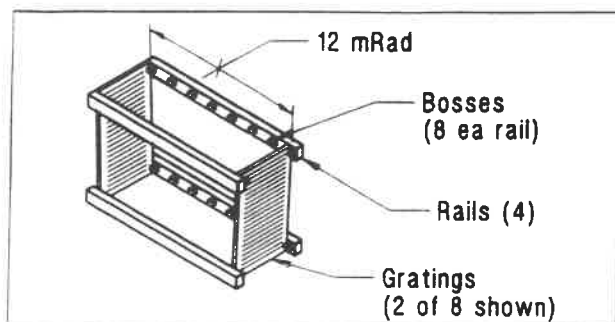


Figure 3. Gratings mounted to rails.

Preliminary Machining

Our first attempt was to use strips from a piece of 0.5" thick cold-worked invar plate. Due to the high length-to-thickness ratio of the rails (18:1), and the residual stresses in the cold-worked material, it proved difficult to machine the rails to the 0.0002" flatness accuracy required for the parts. Our solution was to use a wire electric discharge machine (EDM) to cut the longitudinal profile of each rail from the core of a piece of 1" diameter rough-forged round bar stock, and to modify the annealing heat treatment for the material. The profile of each boss was cut using the wire EDM machine, and then the two rail mounting surfaces were free-abrasive lapped to within 0.0002" flatness and perpendicularity to each other.

Diamond-Turning the Bosses

The geometry of the eight cylindrical grating mounting surfaces on a rail was suitable for plunge-cutting with a flycutter on a 2-axis lathe. The CBN machine at LLNL, a modified Excello model 920 T-base diamond-turning machine, had the correct geometry and slide travel capacities for flycutting these surfaces. Figure 4 is a schematic of the flycutting set-up. A spindle mounted on a slide that travels parallel to the spindle centerline (z-axis) carried a single-point diamond tool set at a 12" swing radius to form the flycutter. A cross-slide that travels perpendicular to the spindle centerline (x-axis) carried the rail. The rail

was mounted on a custom built angle plate that used an angle measuring interferometer for precisely measuring the angle of the rail in the xy-plane of the CBN machine. This allowed the bosses to be flycut so that the normal direction of the mounting surfaces on the bosses, at the rail centerline, could be made to sweep through the necessary 12 mrad angle from the first to eighth boss (Figure 5). By indexing the x-axis relative to the spindle centerline, each boss was precisely positioned so that the flycutter could be plunged into the workpiece to form the 12" cylindrical radius feature on the boss.

The CBN machine uses laser interferometers operating in evacuated beam paths for displacement measurements of the x- and z-axis slides. Abbe off-set errors in indexing the rail relative to the spindle centerline were avoided by mounting the rail in-line with the x-axis measurement path. The spindle uses a hydrostatic bearing that has temperature controlled cooling water to stabilize its thermal growth. Thermally induced dimensional changes in the machine and fixtures are minimized by controlling the temperature of the surrounding air to within ± 0.25 °F.

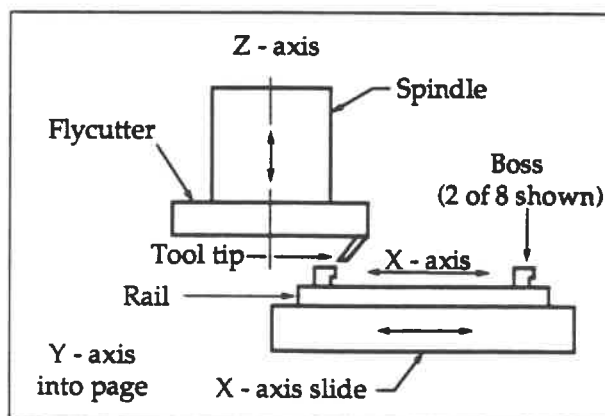


Figure 4. Flycutting set-up.

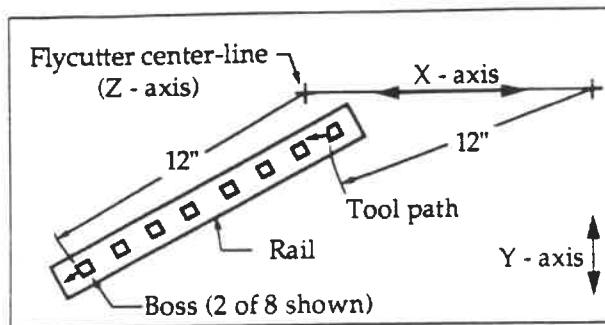


Figure 5. Flycutter tool path and rail set-up.

Qualifying the CBN Machine

To qualify the CBN machine for diamond-turning the rails, a static drift check and a dynamic (machining) test were performed. A static drift test was used to ensure that the machine's laser interferometers, positioning servos, and thermal control were adequate for achieving positioning accuracy on the order of $5\text{ }\mu\text{m}$. Both the x- and z-axis slides were activated and commanded to maintain a static position. Since the positioning accuracy of the x-axis slide was critical for machining the rails, a linear variable differential transformer (LVDT) was set up to measure displacements in the x-direction of the x-axis slide relative to the z-axis slide. Drift of the x-axis slide over 30 minutes was less than $2\text{ }\mu\text{m}$, with a noise level (AC component) of less than $1\text{ }\mu\text{m}$. These results were promising, since the time to sequentially perform the final cuts on all eight bosses on a rail was estimated to be 20 minutes. To further qualify the CBN machine, a qualification part was diamond-turned on it. A circular disc with four radial rows having ten bosses each (forming ten concentric rings) was mounted on the spindle. The disk was made of Ni-Fe Alloy 42, and the bosses were plated with 0.005'' of copper. A single-point diamond tool was mounted to the x-axis slide. Although this arrangement was the opposite of the flycutting set-up intended for the rails, it allowed us to investigate the positioning accuracy of the x-axis slide while the machine was operating in a dynamic cutting mode. The Large Optics Diamond Turning

Machine (LODTM) at LLNL was used as a measuring machine to inspect the location accuracy of the diamond turned surfaces on the qualification part^a. Inspection of the four radial rows of bosses on the part indicated that the positioning error of the CBN machine was within $\pm 8.7\text{ }\mu\text{m}$, which met the $\pm 9.3\text{ }\mu\text{m}$ allowed by our error budget. The qualification part also proved that tool wear due to interrupted cuts would not be an issue for flycutting the eight bosses on a rail.

Error Budget

The error budget for the fabrication and inspection of the boss locations divided the $\pm 15\text{ }\mu\text{m}$ limit on the location error of each boss (e_{total}) among the following error sources: positioning uncertainty of the diamond turning machine (e_D); uncertainty of the angle between the rail center-line and the xz-plane (e_θ); uncertainty of the flycutter radius (e_R); difference in ambient temperature^b during the fabrication of one rail compared to the others (e_T); and the uncertainty of the inspection process for measuring the dimensional accuracy of the finished rails (e_I). After measuring e_D and e_T , reasonable values were assigned for e_θ and e_R , and a maximum permissible value for e_I was calculated. Table 1 shows the sensitivity of the boss location to each of the error sources, and how the $\pm 15\text{ }\mu\text{m}$ was allocated among the error sources. Realizing that the worse case algebraic addition of the errors was probably too conservative, and the root of the sum of the squares (RSS) was probably too optimistic, the average values of the algebraic and RSS allowable errors for each source was used.

^a The measurement repeatability of LODTM was measured to be within $0.2\text{ }\mu\text{m}$.

^b A small ($\approx 10\text{ }\mu\text{m/in}$) consistent scaling up or down of the boss spacing for all four rails would not degrade the optical performance of the grating array.

Inspection results^c

Table 2 shows the measured location errors of the grating mounting surfaces on the four rails. The errors for all four rails were within $\pm 11 \mu\text{in}$. The error span for the first three rails indicated that a progressive degradation in the diamond-turning process was occurring. We suspected that the diamond tool was beginning to exhibit appreciable wear, so it was replaced with a new tool before machining the fourth rail. The $\pm 3.1 \mu\text{in}$ span on the boss locations for the fourth rail verified those suspicions.

At the time of this writing, preliminary optical tests using a Fizeau interferometer and a precision rotary table indicated that the as-mounted orientations of the eight gratings mounted against the bosses of the four rails (EOBB array) were better than what was predicted for the case where the corners of each grating had a $\pm 15 \mu\text{in}$

location error. The final proof of the accuracy of the boss locations will occur in October 1993, when the X-ray performance of the EOBB array will be tested in Europe.

Conclusion

The CBN machine—a T-base diamond-turning machine at LLNL—was successfully used in a flycutting set-up to single-point diamond-turn the grating mounting surfaces of the four attachment rails used for mounting eight X-ray reflection gratings into an array. Each rail has eight grating mounting surfaces that were machined to a position accuracy of $\pm 11 \mu\text{in}$ or better, meeting the required $\pm 15 \mu\text{in}$ tolerance for the parts.

Acknowledgments

Special appreciation goes to Todd Decker, Bob Donaldson, Paul Geraghty, Layton Hale, Maggie Jong, Steve Little, and Greg Wilkinson, all of LLNL.

^c See companion paper: *Inspection of the Diamond-Turned Surfaces Used for Mounting an Array of Eight X-ray Reflection Gratings*, R.C. Montesanti, ASPE 1993 Annual Meeting Proceedings.

Error source	Algebraic addition		Root-Sum-Squares		Ave of RSS & Algebraic Allow errors
	Allowable error	Contribution to e_{total}	Allowable error	Contribution to e_{total}	
e_{θ}	$\pm 3 \text{ arc-sec}$	$\pm 1.0 \mu\text{in}$	$\pm 3 \text{ arc-sec}$	$\pm 1.0 \mu\text{in}$	$\pm 3 \text{ arc-sec}$
e_R	$\pm 0.002''$	$\pm 1.5 \mu\text{in}$	$\pm 0.002''$	$\pm 1.5 \mu\text{in}$	$\pm 0.002''$
e_T	$\pm 0.25 ^\circ\text{F}$	$\pm 2.0 \mu\text{in}$	$\pm 0.25 ^\circ\text{F}$	$\pm 2.0 \mu\text{in}$	$\pm 0.25 ^\circ\text{F}$
e_D	$\pm 6.3 \mu\text{in}$	$\pm 6.3 \mu\text{in}$	$\pm 12.3 \mu\text{in}$	$\pm 12.3 \mu\text{in}$	$\pm 9.3 \mu\text{in}$
e_I	$\pm 4.2 \mu\text{in}$	$\pm 4.2 \mu\text{in}$	$\pm 8.2 \mu\text{in}$	$\pm 8.2 \mu\text{in}$	$\pm 6.2 \mu\text{in}$
e_{total}	—	$\pm 15 \mu\text{in}$	—	$\pm 15 \mu\text{in}$	—

Table 1. Error budget for the fabrication and inspection of the boss locations.

	Boss 1	Boss 2	Boss 3	Boss 4	Boss 5	Boss 6	Boss 7	Boss 8	Span
Rail 1	+ 4.0	+ 4.8	+ 1.2	- 2.5	- 2.7	- 4.8	- 1.0	- 1.2	± 4.8
Rail 2	- 4.9	- 7.8	- 7.1	- 3.0	+ 5.7	+ 7.8	+ 1.9	- 2.9	± 7.8
Rail 3	- 11.0	- 6.7	- 3.9	+ 4.9	+ 11.0	- 2.5	- 2.6	- 1.3	± 11.0
Rail 4	- 0.6	+ 3.1	- 0.2	- 1.5	- 1.7	- 1.0	+ 0.7	- 3.1	± 3.1

Table 2. Location errors of the grating mounting surfaces (μin).

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FUNDAMENTAL STUDY ON MECHANICS OF ELLIPTICAL VIBRATION CUTTING

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SUMMARY

The paper presents a new cutting method named "Elliptical Vibration Cutting", which applies two-directional vibration to the cutting edge in the plane including the cutting direction and the chip flowing direction, so that the chip is pulled out and formed mainly while the tool moves in the chip flowing direction. The tool is restored to the cutting point without cutting while it is separated from the chip. This cutting technique results in remarkable reduction of chip thickness and also cutting force.

1. INTRODUCTION

The ultrasonic vibration cutting has been successfully applied to ultraprecision diamond cutting of difficult-to-cut materials, including steels and glass materials [1]. In the conventional vibration cutting, vibration is applied to the cutting tool only in one direction. Generally, the tool is vibrated in the cutting direction so that the tool is separated from the chip in each cycle of vibration.

The present research presents a new vibration cutting method named "Elliptical Vibration Cutting". Two-directional vibration is applied to the cutting edge in the plane including the cutting direction and the chip flowing direction in such a way that the cutting edge forms an elliptical locus in one cycle of vibration. It is considered that this method has various advantages such as low cutting force, accurate machining, reduction of damage of the cut surface, restraint of chatter vibration and so on, as compared with the ordinary cutting and the conventional vibration cutting.

2. CUTTING PRINCIPLE

Figure 1 shows the principle of the elliptical vibration cutting schematically. In the present method, elliptical vibration is applied to the cutting edge in the plane including the cutting direction and the chip flowing direction, so that the chip is pulled up and formed mainly while the tool moves up in the chip flowing direction. The tool moves down without cutting while it is separated from the chip. Therefore, the relative frictional direction between the tool rake and the chip can be reversed as compared with the conventional cutting methods, and the average frictional force can be less than zero virtually. This virtual lubrication effect increases the shear angle significantly and also reduces the cutting force.

3. EXPERIMENTAL APPARATUS AND CONDITIONS

Figure 2 shows an orthogonal cutting machine equipped with an elliptically vibrated tool developed. The cutting machine is installed within a scanning electron microscope (SEM) in order to observe the cutting processes directly. The tool is vibrated by the two piezoelectric actuators (PZT), and the cutting forces are measured with a piezoelectric type dynamometer. Cutting experiments are carried out by comparing three different cutting methods, which are the elliptical vibration cutting, the conventional vibration cutting and the ordinary cutting.

The tool is made of single crystal diamond. Its rake angle and relief angle are 0° and 7° respectively. Workpiece material is OFC (Oxygen Free Copper) with a thickness of 0.25 mm and a length of 5 mm in the cutting direction. The depth of cut is

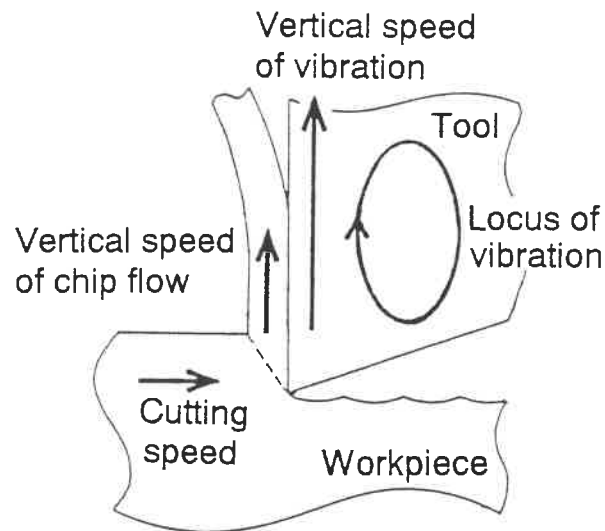


Fig.1 Principle of elliptical vibration cutting.

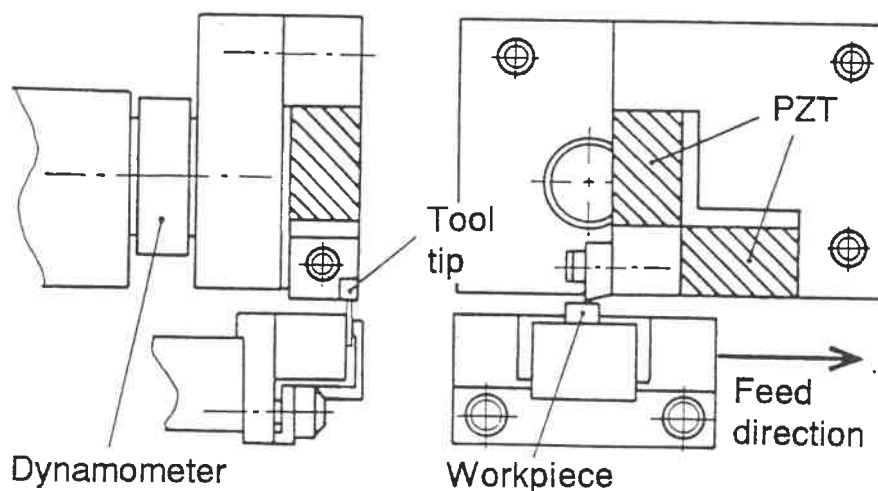


Fig.2 Orthogonal cutting machine equipped with elliptically vibrated tool.

5 μm and the nominal cutting speed, which is equal to feeding speed of workpiece, is 130 $\mu\text{m}/\text{min}$. The circular vibration with a radius of 2.5 μm is applied in the elliptical vibration cutting, while the vibration with an amplitude of 2.5 μm is applied nearly in the cutting direction in the conventional cutting. The vibration frequency is about 1.2 Hz.

Although the cutting speed and the vibration frequency are extremely low and the cutting experiments are carried out in vacuum, it is considered that the basic characteristics of the cutting methods can be clarified.

4. EXPERIMENTAL RESULTS

SEM photographs of the formed chips are shown in Fig. 3. The chip thickness obtained by the conventional vibration cutting is almost the same as that of the ordinary cutting, but that of the elliptical vibration cutting is much less than those obtained by the other methods. This shows that friction on the rake face and also cutting force are reduced significantly by applying the elliptical vibration.

Figure 4 compares the principal and the thrust components of cutting force measured. The arrows in the figures indicate the instances when the tool is located at the leftward end in Figs. 1 and 3. In the conventional vibration cutting, the average cutting force is reduced as compared with the ordinary cutting, because of intermittent cutting, but the cutting force measured during cutting is almost the same. On the other hand, the cutting force is much reduced in the elliptical vibration cutting, even when the tool is cutting. There exists a period when the thrust force is less than zero, which is due to the motion of the tool moving in the chip flowing direction.

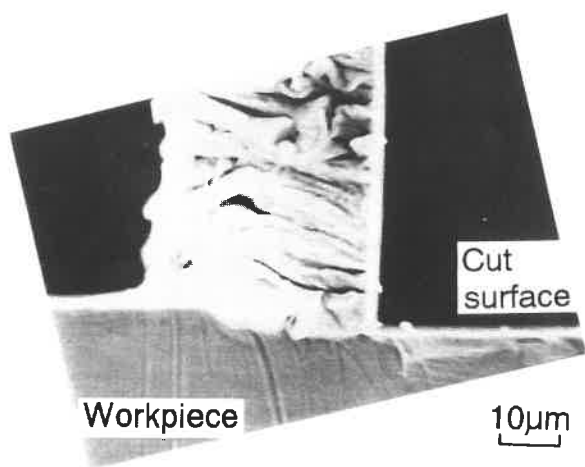
The elliptically vibrated tool generates the finished surface at an instance when it is penetrated into the workpiece. The theoretical roughness calculated numerically is 0.13 μm and the measured roughness is about 0.1 μm under the present cutting conditions. It is predicted theoretically that the surface roughness can be less than 0.1 μm even if this technique is applied at practical cutting speed with an ultrasonically vibrated tool.

5. CONCLUSIONS

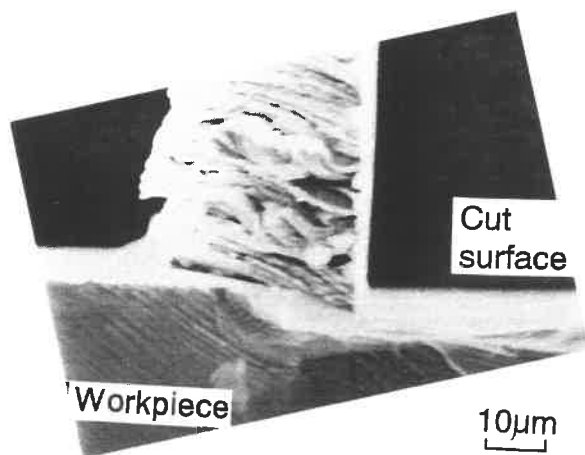
A new cutting method named "Elliptical Vibration Cutting" is proposed in the present paper, and it is confirmed to have the following characteristics. The chip thickness is reduced remarkably as compared with the conventional vibration cutting and the ordinary cutting, and the cutting force is also reduced especially in the thrust direction. It is expected that these effects lead to various advantages such as improvement of machining accuracy, reduction of damage of the cut surface, restraint of chatter vibration and cutting of difficult-to-machine materials.

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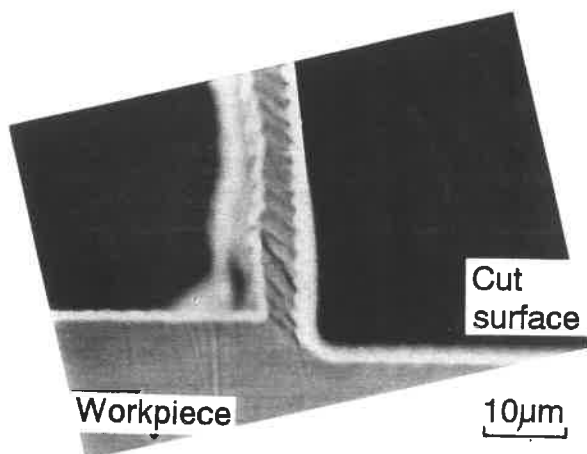
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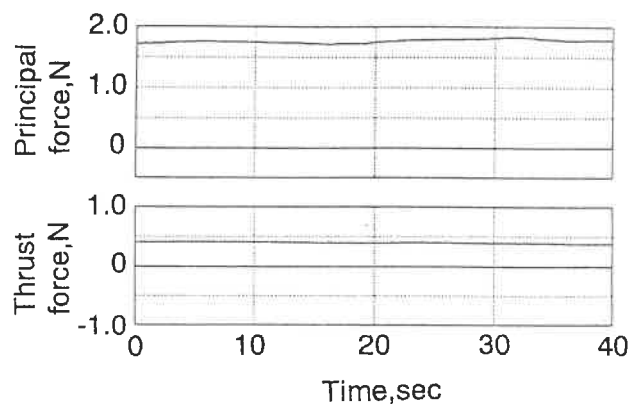
(a) Ordinary cutting.



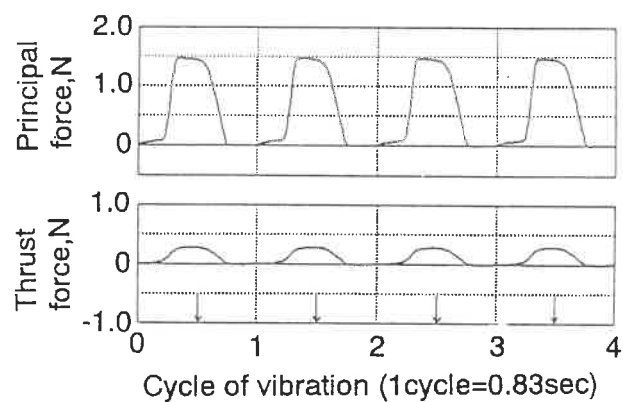
(b) Conventional vibration cutting.



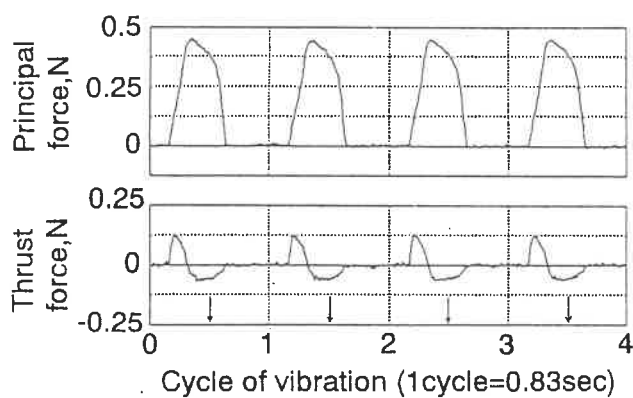
(c) Elliptical vibration cutting.



(a) Ordinary cutting.



(b) Conventional vibration cutting.



(c) Elliptical vibration cutting.

Fig.3 SEM photographs of formed chips.

Fig.4 Principal and thrust components of cutting force measured.

Optical Finishing of Quartz Crystal by Diamond Turning

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Abstract

The possibility of producing fine optical surfaces in single-crystal quartz was examined through diamond turning with a highly stiff and accurate machine tool. The effects of depth of cut, feed rate, and cutting speed on surface features, roughness, flatness, and residual strain were investigated. The best surface obtained was optically transparent with a surface roughness better than $2 \text{ nmR}_{\text{max}}$, and negligibly small residual strain.

1. Introduction

Quartz crystals for generating high frequencies are increasingly in demand for computers, communications, and so on^{1),2)}. In the case of a 100 MHz resonator, an AT-cut quartz crystal with a thickness of only about $17 \mu\text{m}$ is required. However, it is impossible to produce such a very thin quartz crystal by conventional lapping and polishing. Recently, the production of optical quality surfaces on brittle materials such as silicon, germanium, and glass using ultra-precision machine tools has been studied^{3),4)}. We have studied the applicability of diamond turning to the fabrication of a thin quartz wafer, which is one of most difficult materials for turning because it is harder and more highly brittle than the other materials. For this purpose, we constructed a highly stiff and accurate machine tool.

This paper describes that the relationship of the features of the finished surfaces to cutting conditions such as depth of cut, feed rate, and cutting speed was examined. In addition, residual strain caused by turning was evaluated by double-crystal x-ray diffraction.

2. Experimental

A high performance machine tool, composed of an air-bearing spindle with an axial dynamic stiffness of $20 \text{ N}/\mu\text{m}$ and an air slide with a dynamic stiffness of $250 \text{ N}/\mu\text{m}$ perpendicular to the axis of motion, was used. The diamond toolholder was driven by a piezoelectric actuator with a resolution of less than 1 nm to control the depth of cut. Two types of toolholder with a static stiffness of $5 \text{ N}/\mu\text{m}$ and $24 \text{ N}/\mu\text{m}$ in parallel with the axis of motion giving depth of cut were used. The natural diamond cutting tool used had a rake angle of -20 deg. , a clearance angle of 10 deg. , and a tool-nose radius of 0.8 mm . Polished AT-cut quartz wafers of 12.5 mm diameter and 1 mm thickness were fastened on the spindle and turned. Finished surfaces were observed using a Nomarski microscope, transmission electron microscope (TEM), and an atomic force microscope (AFM). Surface profiles were measured with a Talystep profilometer. The flatness of the turned surfaces was measured using a laser interferometer. The residual strain in the turned surface layer was evaluated using the double-crystal x-ray diffraction method.

3. Results and Discussion

3.1 Surface features

Fig.1 is Nomarski micrographs and surface profiles showing the effect of the depth of cut on the turned surfaces. The feed rate and mean cutting speed were $8.1 \mu\text{m}/\text{rev}$ and 3.0 m/s , respectively. At 500 nm cutting depth, an optically clear surface was not obtained. The surface texture was composed of microcracks and the surface roughness was about $900 \text{ nmR}_{\text{max}}$ as shown in Fig.1(a). At 300 nm cutting depth, a transparent surface with a roughness of about $11 \text{ nmR}_{\text{max}}$ was obtained (Fig.1(b)). Grooves in parallel with the cutting direction were observed. The $8 \mu\text{m}$ spacing of the grooves corresponds to the feed rate. Furthermore, shallower grooves, attributable to the imperfect

cutting edge were also observed. As the depth of cut was decreased to 100 nm, the surface roughness improved to about 6 nmR_{max} (Fig.1(c)).

Fig.2 shows Nomarski micrographs and surface profiles of the turned surfaces made using a toolholder with a higher static stiffness of 24 N/ μ m. The surface roughness was about 4 nmR_{max} and about 2 nmR_{max} at 300 and 100 nm cutting depth, respectively. Comparing this to Figs.1(b) and (c), the surface roughness obtained using the toolholder with a higher static stiffness was better than that obtained with a lower one at the same depth of cut. At a depth of cut of 300 nm, grooves corresponding to the feed rate (similar to Fig.1(b)) were observed as shown in Fig.2(a). As the depth of cut was decreased to 100 nm, a transparent surface with no groove on the turned surface was obtained (Fig.2(b)). This surface feature is quite comparable to commercial polished surfaces.

Fig.3 is a TEM image of the replica of the turned surface shown in Fig.2(a). Plastic deformation and flow without microcracks were observed on both sides of the groove.

Fig.4 shows a SEM photograph of typical continuous chips obtained by turning at 100 nm cutting depth. On their outer surface, a lamellar structure is seen. The chip morphology resembles that obtained in metal turning. The transparent surfaces, plastic deformation, and continuous chip formation prove that ductile-regime turning was achieved.

Fig.5 is Nomarski micrographs showing the effects of the feed rate and cutting speed on turned surfaces at 4.1 μ m/rev feed rate and 300 nm cutting depth. The mean cutting speeds were 3.0 and 6.0 m/s in Figs.5(a) and (b), respectively. A surface roughness of about 2 nmR_{max} at 3.0 m/s mean cutting speed and about 1 nmR_{max} at 6.0 m/s was obtained. At the same depth of cut at 300 nm, decreasing the feed rate from 8.1 μ m/rev to 4.1 μ m/rev resulted in an improved surface roughness of less than 2 nmR_{max}. However, the effect of cutting speed was not obvious under these conditions.

3.2 Residual strain

Fig.6 is an x-ray topograph of the (02 $\bar{2}$ 2) reflection of the turned surface shown in Fig.5(a). In this topograph, contrasts along the cutting direction were observed. This indicates that strain exists in the turned surface layer along the grooves observed on the surface. X-ray topographs measured from most of the turned surfaces obtained under different cutting conditions were similar to that shown in Fig.6.

Fig.7 is an x-ray topograph of the turned surface shown in Fig.2(b). Sharp contrasts along the cutting direction, as seen in Fig.6, were not observed. This suggests that strain exists only in an extremely thin layer. Topographs like this, which resemble that of a polished surface, were obtained only at a depth of cut of 100 nm.

Fig.8 shows a typical three-dimensional profile of a turned surface (shown in Fig.5(a)) measured by laser interferometer. The flatness was as high as 193 nm peak-to-valley at about 10 mm diameter, but the variation in the thickness of the sample was as uniform as 18 nm peak-to-valley. This datum indicates that the sample bent in a convex shape towards the turned surface due to residual strain. Most of the samples turned under different cutting conditions were convex as shown in Fig.8.

The residual strain in the turned surfaces was evaluated from an x-ray rocking curve measurement of the (01 $\bar{1}$ 1) reflection. The full-width at the half-maximum (FWHM) values of the rocking curves indicates the amount of strain at the surface. Decreasing the depth of cut from 300 to 100 nm resulted in a decreased FWHM value, i.e., the residual strain in the turned surface decreased as the depth of cut decreased. The residual strain in the turned sample at a depth of cut of 100 nm was as small as that for a conventionally polished one. This result indicates that residual strain was negligibly small. The residual strain also decreased as the feed rate decreased at the same depth of cut. In contrast, under the same cutting conditions, the residual strain using the toolholder with a higher static stiffness was smaller than that with a lower one.

Fig.9 shows the relationship of FWHM value to flatness. Table 1 summarizes the cutting conditions corresponding "A~F" in Fig.9 and polishing is shown by "P". Although the data are scattered, flatness shows a tendency to be proportional to FWHM value, i.e., the convex shape becomes more remarkable as the residual strain increases. At a depth of cut of 100 nm, the flatness was 45 nm peak-to-valley.

Fig.10 shows the relationship of FWHM value to surface roughness measured with the Talystep and AFM. "A~F" and "P" in Fig.10 are similar to those in Fig.9. In the cases of "A" and "P", the

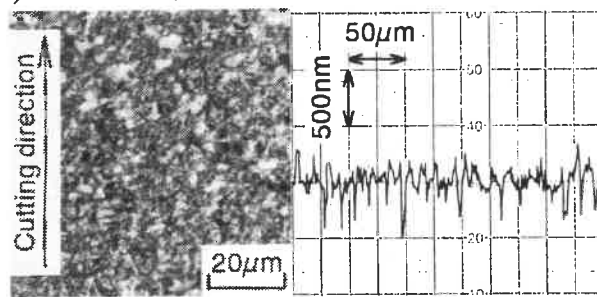
surface roughness values coincided with those obtained with the AFM. On the other hand, the surface roughness values obtained with the AFM are greater than those with the Talystep except for the above two cases, because a Talystep stylus can not trace the bottom of a deep groove accurately. In both cases, however, surface roughness increased lineally as residual strain increased.

4. Conclusions

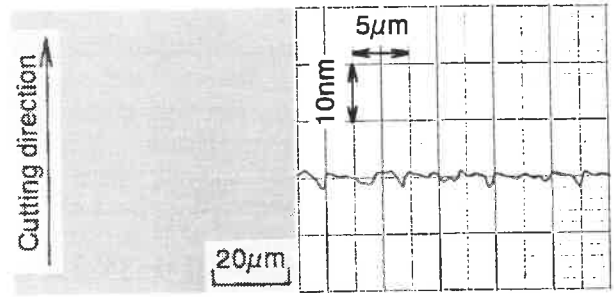
Surface roughness and residual strain decreased with decreasing depth of cut and/or feed rate. Increasing the static stiffness of a toolholder shows the considerable effect to improve surface roughness and residual strain. The best surface obtained was optically transparent with a surface roughness less than 2 nm_{Rmax}. The residual strain was negligible, and consequently, the flatness was 45 nm. These results prove the possibility of producing fine optical surfaces in quartz by using a machine tool with a high stiffness and a tool positioning control at the nanometer level.

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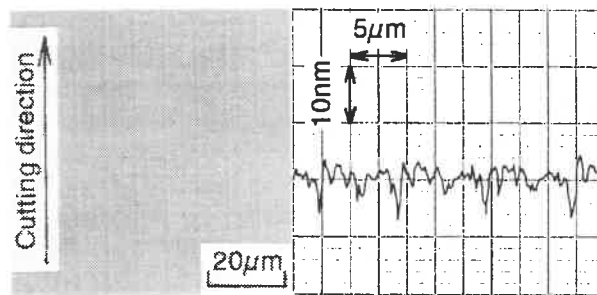
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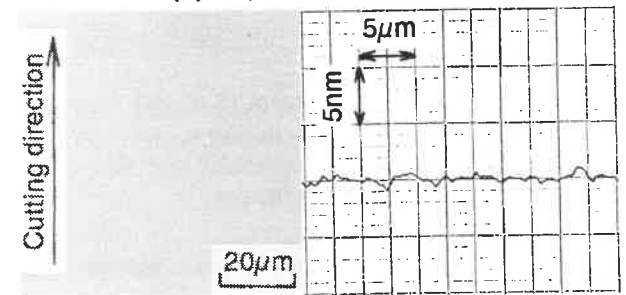
(a) depth of cut : 500nm



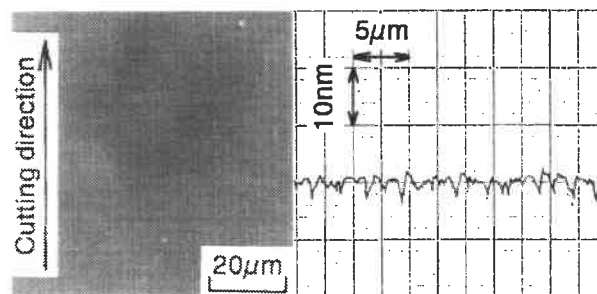
(a) depth of cut : 300nm



(b) depth of cut : 300nm



(b) depth of cut : 100nm



(c) depth of cut : 100nm

Fig.1 Nomarski micrographs and surface profiles with Talystep profilometer of turned surfaces (feed rate : 8.1μm/rev, mean cutting speed : 3.0m/s, stiffness of toolholder : 5N/μm)

Fig.2 Nomarski micrographs and surface profiles with Talystep profilometer of turned surfaces (feed rate : 8.1μm/rev, mean cutting speed : 3.0m/s, stiffness of toolholder : 24N/μm)

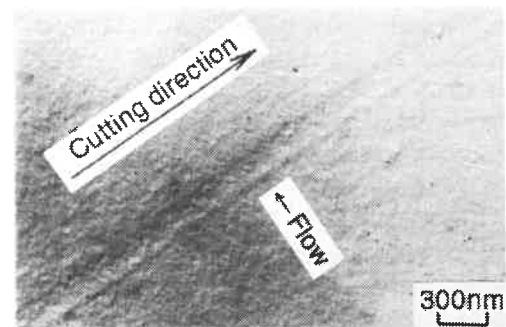


Fig.3 TEM image of replica of turned surface showing plastic deformation and flow (depth of cut : 300nm, feed rate : 8.1 μm/rev, mean cutting speed : 3.0m/s)

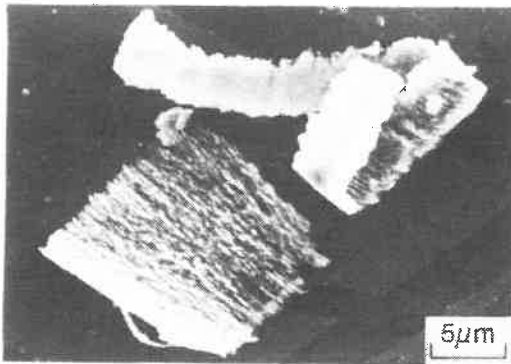
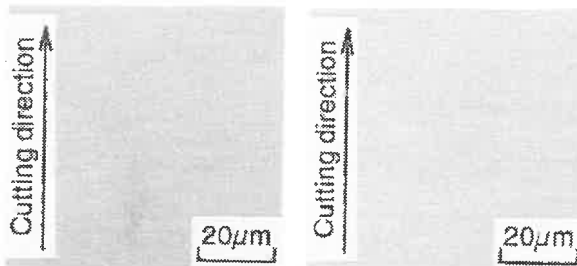


Fig.4 SEM photograph of typical continuous chips obtained from turning quartz crystal (depth of cut : 100nm, feed rate : 8.1 $\mu\text{m}/\text{rev}$, mean cutting speed : 3.0m/s)



(a) depth of cut : 300nm feed rate : 4.1 $\mu\text{m}/\text{rev}$ mean cutting speed : 3.0m/s
(b) depth of cut : 300nm feed rate : 4.1 $\mu\text{m}/\text{rev}$ mean cutting speed : 6.0m/s

Fig.5 Nomarski micrographs of turned surfaces

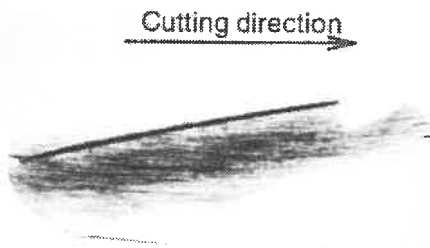


Fig.6 X-ray topograph of typical turned surface (depth of cut : 300nm, feed rate : 4.1 $\mu\text{m}/\text{rev}$, mean cutting speed : 3.0m/s)

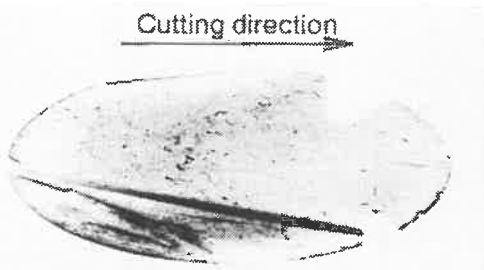


Fig.7 X-ray topograph of turned surface at 100 nm cutting depth (feed rate : 8.1 $\mu\text{m}/\text{rev}$, mean cutting speed : 3.0m/s)

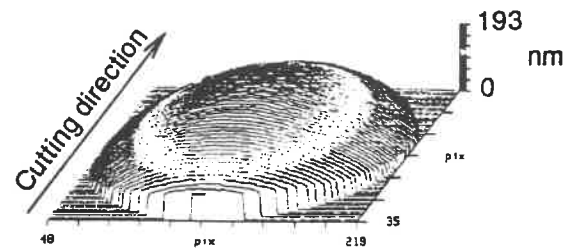


Fig.8 Typical three-dimensional profile of turned surface with laser interferometer (depth of cut : 300nm, feed rate : 4.1 $\mu\text{m}/\text{rev}$, mean cutting speed : 3.0m/s)

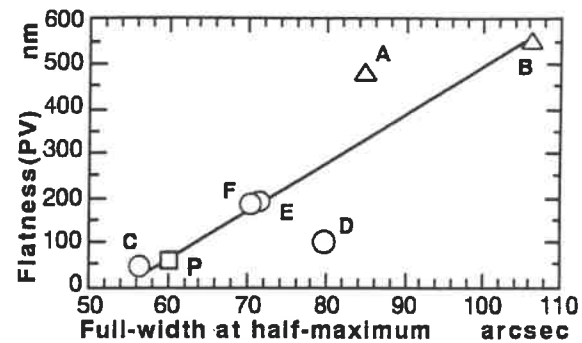


Fig.9 Relationship of FWHM value to flatness ("A~F" : turned surfaces (Table 1), "P" : polished surface)

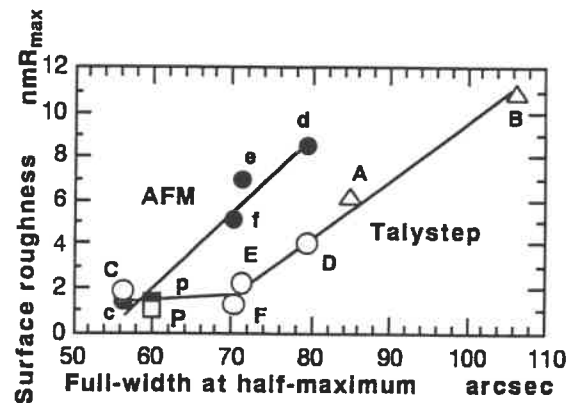


Fig.10 Relationship of FWHM value to surface roughness with Talystep and AFM ("A~F" : turned surfaces (Table 1), "P" : polished surface)

Table 1 Summary of cutting conditions of "A~F" plotted in Fig.9 and Fig.10

	A	B	C	D	E	F
depth of cut (nm)	100	300	100	300	300	300
feed rate ($\mu\text{m}/\text{rev}$)	8.1	8.1	8.1	8.1	4.1	4.1
mean cutting speed (m/s)	3.0	3.0	3.0	3.0	3.0	6.0
stiffness of toolholder (N/ μm)	5	5	24	24	24	24
symbol	Δ	Δ	$\circ$	$\circ$	$\circ$	$\circ$

THERMAL ENERGY FROM MACHINING PROCESS

A Dominating Factor on Form Accuracy in Ultra- and High-Precision-Turning

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For flexible production of high and ultra-precision parts, turning is a valuable technology to achieve highest surface quality even in comparison with other techniques like grinding and polishing. The essential goal is to achieve best form and dimensional accuracy. Within the last years lots of effort was spent into the optimization of machine tools. Static and dynamic stiffness have been brought to a point that best form accuracies could be expected. Also the thermal behavior of these machines was improved essentially. Anyway, the results in production are not as well as they should be.

For this reason the thermal influence of the cutting heat in diamond turning and precision-turning of hardened steel was analysed by temperature measurements inside the tool and the workpiece. After each machining process the form deviation was analysed. A direct relation between temperature rise in the tool, tool-shaft and workpiece during cutting process and the machined form accuracy of the machined part was found.

During precision-turning of hardened steel the temperature changes were detected to be in between 5 to 15 K, depending on the process parameters, and in the same order of magnitude in tool shaft and workpiece. By using a capacitive probe the machined contour of the test-workpiece was measured directly on the machine. With this method any influence from an alignment error between spindle and slides can be avoided. In [figure 1](#) the temperature curves of tool and workpiece during process and the measured form deviation are shown. In order to reduce these thermal induced errors a temperature control system for the cooling lubricant was installed. From a temperature stabilized tank the cooling lubricant is sprayed with 2 independent nozzles on the workpiece and the tool. Using this system, a constant temperature (21 °C) during the whole machining process can be expected.

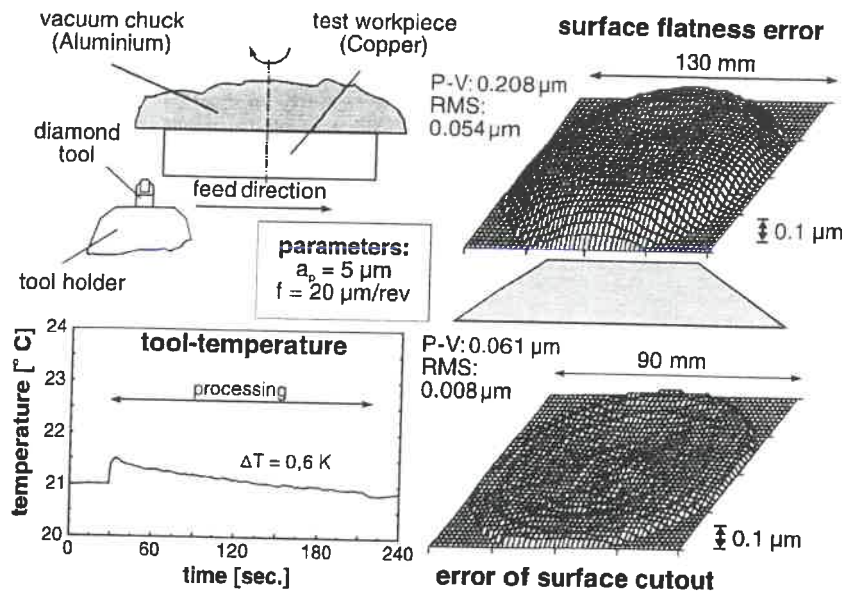


Fig. 5: Temperatures and surface form error with cooling system

were lowered about 50 %. In this case the P-V form-deviations could be reduced also by nearly 50 %.

Additionally the type of form error changed from a conical shape to a large flat surface with a steep tapered edge. Although the dependence of the form error on cutting depth maintains because the cooling system cannot avoid the heating up of the tool during start of cutting completely, this means a significant optimization of the diamond turning process. In general the tapered edge is more acceptable than a wire edge. Especially for laser mirrors the outer area has no optical function. Reducing the measured area of the machined test mirror from the original 130 mm diameter to 90 mm (70 %) the form deviation comes down to 61 nm P-V (8 nm RMS), figure 5. This value corresponds to less than $\lambda/100$ of the typical CO_2 -laser wavelength of $10.6 \mu\text{m}$.

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Poster Papers



A S P E

VIBRATION EFFECT ON CUTTING PROCESS USED OD-BLADE

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1. Introduction

Authors developed a new method in order to improve machining efficiency and accuracy on cutting used OD-blade for hard and brittle materials, such as single-crystal silicon and ceramics etc.. This new method is the vibration cutting used OD-blade (OD-blade is a cutting tool fixed diamond grains on the outside of the circular blade). The vibration cutting is the method applied low frequency ($\leq 30\text{Hz}$) and small amplitude for a workpiece, and the vibration direction is parallel to the cutting direction (as shown in Figure 1). When the vibration velocity amplitude $>$ the cutting speed (feed speed of the workpiece), the vibration cutting brings about a phenomenon of cutting and non-cutting during one cycle of vibration. Therefore, as the working fluid enters effectively into the processing parts, it can be expected that cutting force in the vibration cutting is less than one in the non-vibration cutting and the tool life in the vibration cutting is improved to be compared with the non-vibration cutting.

This paper describes about the effect of the vibration cutting used OD-blade. At first, the experimental apparatus and method are explained. After that behavior of cutting force, grinding ratio and cutting accuracy etc. are described in experimental results.

2. Experimental apparatus and method

Figure 2 shows the outline of an experimental apparatus. Vibration exciter for workpiece (as shown in Figure 3) is set on the working table in the apparatus. The equipment is able to change rotation of motor into

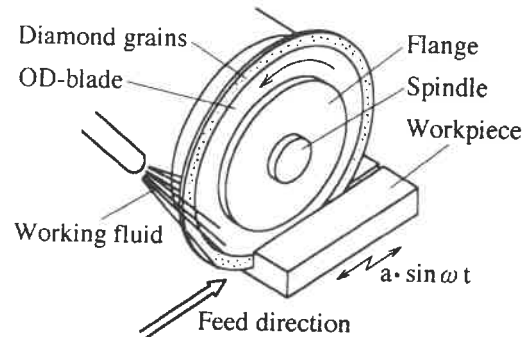


Fig.1 Outline of vibration cutting used OD-blade

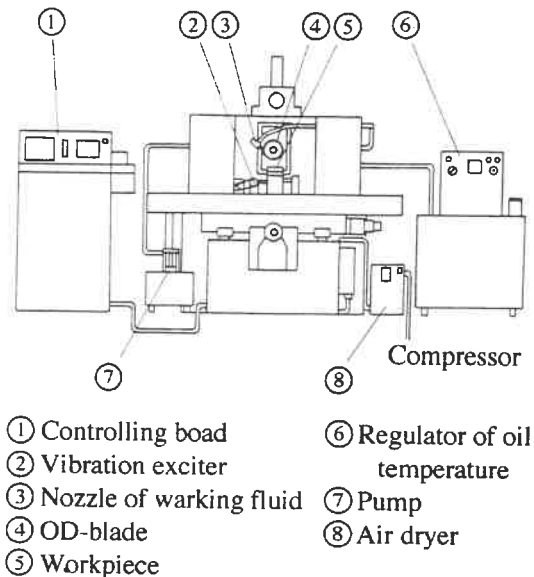


Fig.2 Outline of experimental apparatus

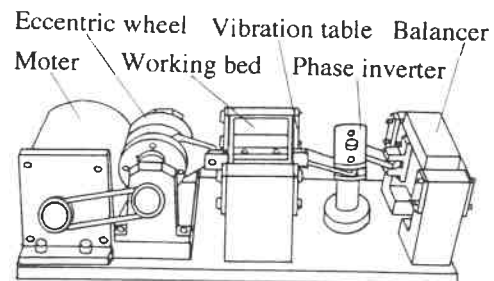


Fig.3 Outline of vibration exciter

Table.1 Experimental conditions

Workpiece	Soda glass $50 \times 60 \times 10^1$ mm
OD-blade	$\phi 125 \times 1$ mm SD#200 Metal bond
Blade speed : V	1915 mm/min
Cutting depth : g	8, 11 mm
Feed : v	20~80 mm/min
Working fluid	Water soluble coolant
Amplitude : a	0, 0.2 mm
Frequency : f	0~28 Hz

reciprocating motion by means of the eccentric wheel and to prevent an inertia force of the work stage during vibration cutting.

In case of this experiment, the workpiece is fixed on the work stage used wax, and the cutting blade is set on the air spindle of the apparatus. The rotating number of OD-blade, feed speed of the table and cutting depth are set up. Working fluid is supplied plentifully on the cutting part. The main experimental conditions are shown in Table 1.

3. Experimental results and considerations

Figure 4 shows observations of cutting groove used a normal blade (symmetric blade) and an asymmetric blade. This study uses these 2 types of the blade.

3.1 Cutting force

Figure 5 shows the relationship between the frequency and motor current for rotation of the blade, and then (a) is results of cutting used the normal blade and (b) is results of cutting used the asymmetric blade. When the frequency increases, it is clear that the motor current decreases. Because, when the frequency increases, cutting amount per one vibration cycle decreases. When the frequency is 28 Hz, the motor current of normal blade is 50 % compared with the motor current in non-vibration cutting. The motor current of asymmetric blade becomes about 60 %, and the motor current in the vibration cutting shows

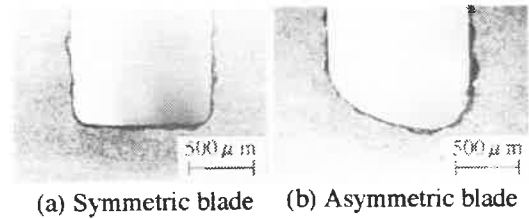


Fig.4 Observations of cutting groove

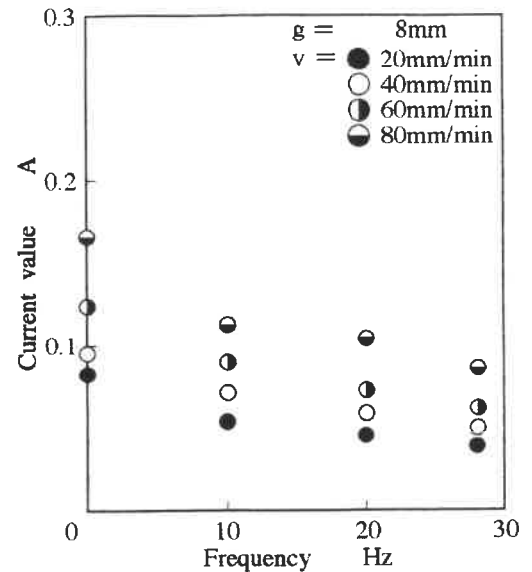


Fig.5 (a) Relationship between frequency and current value (symmetric blade)

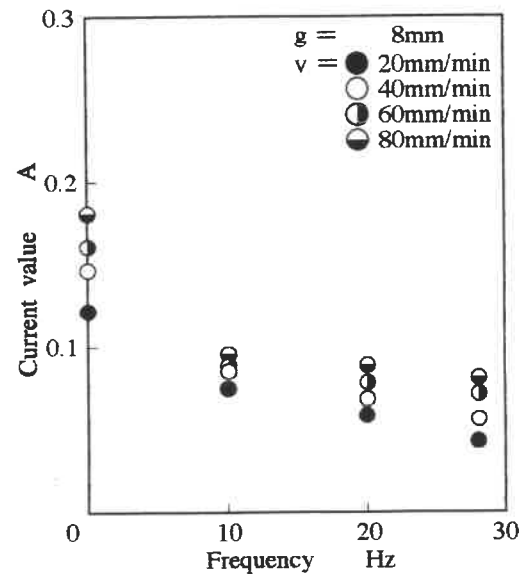


Fig.5 (b) Relationship between frequency and current value (asymmetric blade)

same value as the normal blade in the non-vibration cutting. In the non-vibration cutting, the cutting force at asymmetric blade acts for the direction of blade width, and then motor current increased. However, the vibration cutting brings about phenomena of cutting and non-cutting during one cycle of vibration (when the vibration velocity amplitude $>$ the cutting speed). Therefore, diffraction of blade during cutting is decreased and cutting force is decreased.

Next, the cutting force after 150 times of cutting number is examined. The cutting force in the non-vibration cutting shows five times of initial cutting. When frequency is 28 Hz, the cutting force after 150 times shows as same value as it of first time. It is clear that the vibration cutting is difficult to bring about glazing and attritious wear.

3.2 Grinding ratio

Figure 6 shows the relationship between frequency and grinding ratio. The grinding ratio means a grinding amount of workpiece per a wear volume of blade. The grinding ratio at 10 Hz is decreased to be compared with it in non-vibration, and in case of 30 Hz, the grinding ratio is larger than the result of the non-vibration cutting.

Figure 7 shows observations of blade surface after 150 times of slicing. A series of illustrations of the right side in Figure 7 are model of photographs observed. It is clear that many diamond grains on the blade surface in the non-vibration cutting are worn. When frequency is 10 Hz, many diamond grains are fallen from the blade surface. And falling of diamond grains is less than 10 Hz in proportion to increasing of the applied frequency.

3.3 Cutting accuracy

When symmetric and asymmetric blades are used in the slicing process, Figures 8 and 9 show the waviness of the cutting surface. In case of the non-vibration cutting, the surface

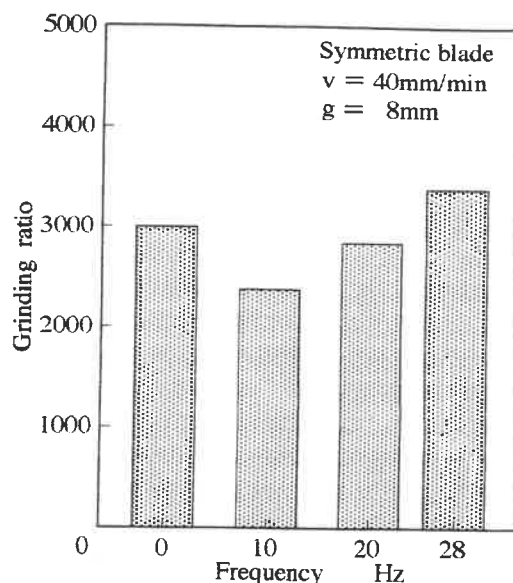


Fig.6 Relationship between frequency and grinding ratio

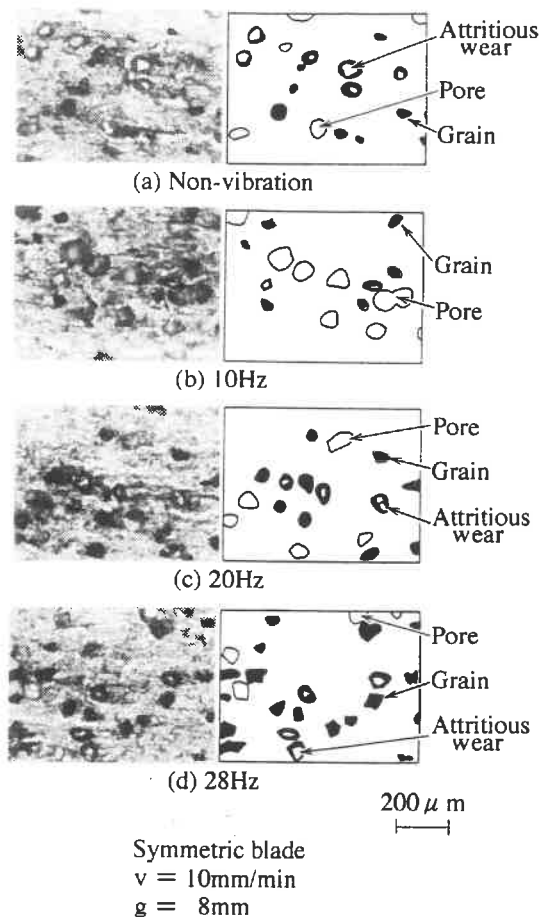


Fig.7 Observations of blade surface after 150 times

waviness used symmetric blade are about 3~4 μm , while the waviness used asymmetric blade shows about 6 μm . When vibration is applied during cutting, the surface waviness used both blades are decreased with increase of frequency. And the surface waviness by the asymmetric blade at about 30 Hz is the same value as it when the symmetric blade is used in the non-vibration cutting.

Figure 10 shows the model of the cutting process used the asymmetric blade. As the asymmetric blade is diagonal at peripheral surface of cutting edge, radial force acts on the cutting blade. Therefore, radial runout of the asymmetric blade increases to be compared with the symmetric blade, and surface waviness used the asymmetric blade deteriorates to be compared with it owing to the symmetric blade. As the vibration cutting brings about phenomena of cutting and non-cutting during one cycle of vibration, the radial force acted on the blade becomes to negligible small in the non-cutting process. Therefore, the vibration cutting improves surface accuracy of the workpiece to correct for radial runout.

5. Conclusions

The results obtained through a series of experiments are as follows:

- (1) The cutting force of the vibration cutting becomes less than one of non-vibration cutting. And the cutting force decreases in proportion to increasing of the applied frequency.
- (2) When the asymmetric wear is observed on a cross-section of blade outer edge, the cutting force is larger than the symmetric blade. But, even if the blade with the asymmetric wear is used in the vibration cutting, the cutting force is approximately equal to that used the symmetric blade.
- (3) Wear volume of the blade in the vibration cutting changes at each frequency. As the frequency increases, the wear volume of blade decreases. And the wear volume of blade in the vibration cutting (≥ 25 Hz) is the same with it in the non-vibration cutting.
- (4) Waviness and thickness of the wafer by means of the vibration cutting is smaller than those of the non-vibration cutting. And this tendency is clear in case of cutting used asymmetric blade.

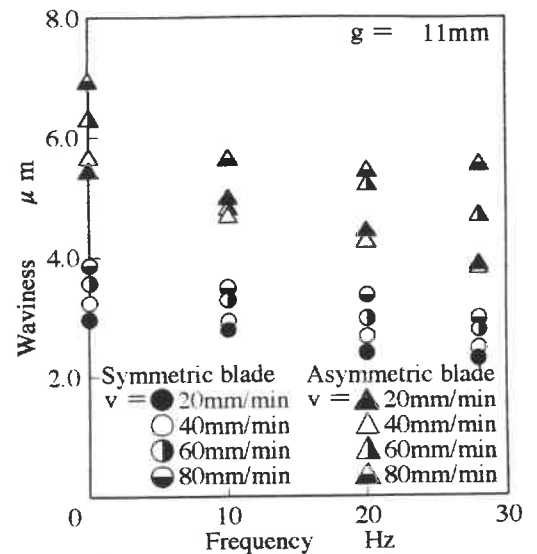


Fig.8 Relationship between frequency and waviness

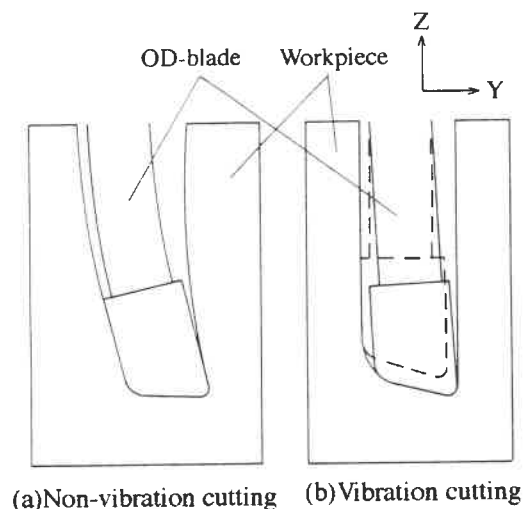


Fig.9 Cutting model by asymmetric blade

EFFECT OF WHEEL CHARACTERISTICS ON DUCTILE REGIME GRINDING OF BRITTLE MATERIALS

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INTRODUCTION

The performance criteria for the machining of brittle materials include figure accuracy, surface finish, and subsurface fracture damage. Grinding is an attractive fabrication method (as compared to diamond turning or polishing) because high figure accuracies are achievable at high material removal rates on a wide variety of materials. In brittle materials, however, grinding induces cracks. Previous researchers have shown that ductile regime grinding which leaves a fracture free surface is possible but that the chip thicknesses required are small--less than 10 nm for glasses and 100-200 nm for tough ceramic materials. The critical chip thickness (d_c) for ductile grinding depends on many factors including workpiece material, grinding geometry, wheel characteristics, and coolant chemistry. Different manufacturers have different "rules of thumb" for selecting a diamond/binder system for a specific workpiece. With a more quantitative selection methodology, wheel systems could be optimized more quickly and effectively. This paper focuses on the relationship between wheel characteristics and the resulting ground surface.

THEORETICAL MODEL

A wheel parameter model has been devised to predict the effect of grit size, diamond concentration, and binder modulus on the critical chip thickness. Based on indentation theory, a model was developed by Bifano to relate the critical depth (or chip thickness) to material parameters[1]:

$$d_c \propto \left(\frac{E}{H}\right) \left(\frac{K_c}{H}\right)^2 \quad (1)$$

E , H , and K_c are the modulus of elasticity, hardness, and fracture toughness, respectively.

The indentation model predicts the critical depth based on a single cutting point. To account for wheel parameter effects, this model has been extended to account for multiple cutting points. In an indentation, $P \propto d^2 H$. The critical load for a single cutting point is therefore:

$$P_c \propto \left(\frac{E}{H}\right)^2 \frac{K_c^4}{H^3} \quad (2)$$

In the multiple cutting point model, the critical load for fracture is the sum of multiple grit loads and is therefore higher than a single indentation load. This model describes the force between wheel and workpiece and relates the critical chip thickness to the summation of the individual grit forces.

Grits in a compliant binder were modeled as spherical masses attached to springs as shown in Figure 1 [2,3]. The sum of individual grit forces yields a total normal force between wheel and workpiece. At the critical depth of cut, this total normal force is:

$$F_{total,cr} = \sum_{i=0}^{N_g} P_i \quad (3)$$

where,

- N_{cr} = number of grits in contact with workpiece when at the critical depth of cut = $N_v''' A_c \delta_{cr}$
- N_v''' = number of grits per volume of abrasive matrix
- A_c = area of the wheel/workpiece contact envelope
- δ_{cr} = wheel deflection at the critical depth of cut
- P_i = force on grit i = $P_i(E_{binder}, r_{grit}, e_i)$
- E_{binder} = modulus of elasticity of wheel binder
- r_{grit} = average radius of abrasive grits
- e_i = deflection of grit i = $i l_m$

l_m = mean distance between grits in the direction of deflection = $l/(N_v''' A_c)$

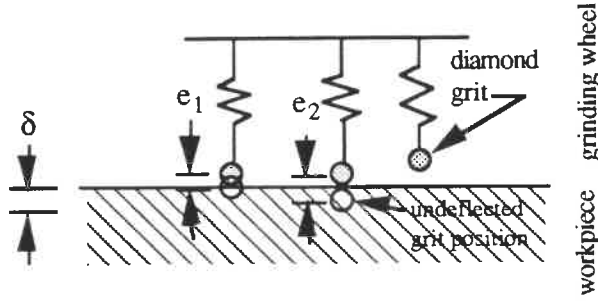


Figure 1: Force model of wheel/workpiece interaction: wheel has been displaced relative to workpiece by δ

Number of Cutting Points per Volume

The volume fraction and the average particle volume determine the number of grits per unit volume. Assuming spherical diamond particles, N_v''' can be expressed as a function of diamond concentration $c_{diamond}$ and grit radius r_{grit} :

$$N_v''' = \frac{3}{16\pi} \frac{c_{diamond}}{r_{grit}^3} \quad (4)$$

Force-Deflection Characteristic Two force-deflection relationships for the grit/binder interface were considered. A Hertzian model for a sphere on a flat gives:

$$P_i = 1.48 E_{binder} r_{grit}^{1/2} e_i^{3/2} \quad (5)$$

Because the diamond particles are embedded in the binder, a relationship which models the force between a rigid circular cylinder and a flat semi-infinite elastic solid may be more appropriate:

$$P_i = \frac{2}{1 - \nu_{binder}^2} E_{binder} r_{grit} e_i \quad (6)$$

Critical Wheel Deflection Calculating δ_{cr} requires a criteria for the ductile-to-brittle transition. The criteria for a single cutting point is that P_i must exceed P_c . For multiple cutting points, the transition is assumed to occur when the force on any one grit exceeds P_c . The critical wheel deflection is equal to the

deflection of the outermost cutting point in response to a load of P_c : $\delta_{cr} = e_{max} = e_{max}(E_{binder}, r_{grit}, P_c)$. Rearranging the force-deflection relationship of Equation (6), the critical wheel deflection is:

$$\delta_{cr} = \frac{1 - \nu_{binder}^2}{2} \frac{P_c}{E_{binder} r_{grit}} \quad (7)$$

Combining Equations (3), (4), (6), and (7), the total normal force at the critical depth can be rewritten as:

$$F_{total,cr} = \frac{P_c}{2} (N_{cr} + 1) \quad \text{where}$$

$$N_{cr} = \frac{3}{32\pi} (1 - \nu_{binder}^2) \frac{c_{diamond}}{E_{binder} r_{grit}^4} P_c A_c \quad (8)$$

If $N_{cr} \gg 1$,

$$\frac{F_{total,cr}}{A_c} = \frac{3}{64\pi} (1 - \nu_{binder}^2) \frac{c_{diamond}}{E_{binder} r_{grit}^4} P_c^2 \quad (9)$$

Research on the grinding of metals (ductile regime grinding) has shown that the normal grinding force is approximately proportional to the depth of cut [4,5], or

$$d_c \propto F_{total,cr} \quad (10)$$

Therefore,

$$d_c \propto \frac{c_{diamond}}{E_{binder} r_{grit}^4} \quad (11)$$

Equation (11) predicts that critical depth of cut increases with increasing diamond concentration, decreasing binder modulus, and decreasing grit size.

EXPERIMENTAL RESULTS

The model summarized in Equations (9) and (10) has some surprising features. Whereas $d_c \propto \sqrt{P_c}$ in the indentation theory, $d_c \propto P_c^2$ in the multiple cutting point model. Also, the inverse dependence on grit size is particularly strong. Cutting tests on a variety of materials using a variety of wheels are necessary to verify the P_c and wheel parameter dependences.

Grinding tests were performed on two materials--soda lime glass and nickel-zinc ferrite--using four metal grinding wheels which varied in grit size and concentration. The tests were conducted on a two axis diamond turning machine with an air turbine grinding motor. The geometry is illustrated in Figure 2. During each test, chip thickness was changed by varying the feedrate (from 0.15 mm/min up to 20 mm/min) while holding the wheel speed constant at 60,000 rpm.

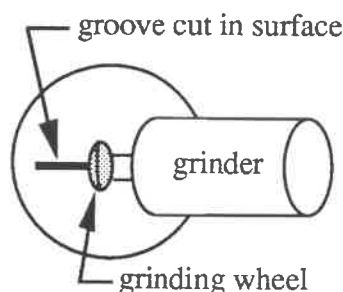


Figure 2: Grinding geometry

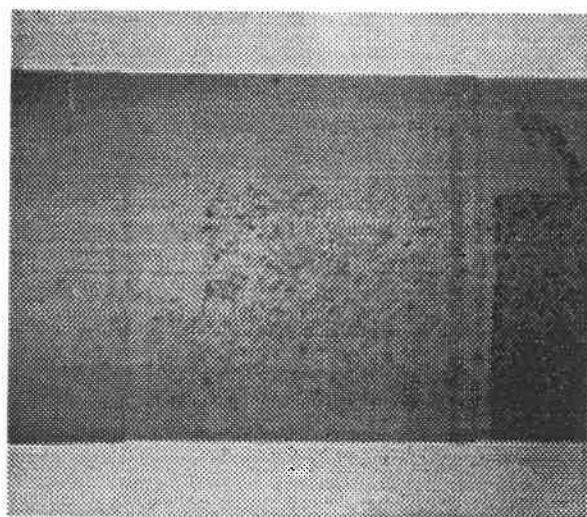


Figure 3: Nomarski micrograph (50X) of "crossfeed" ground soda lime glass showing transition from a ductile to a brittle regime

Figure 3 shows a portion of the groove resulting from one of these tests. It shows a transition from an unfractured region (at far left) ground at 0.15 mm/min to a moderately fractured region (center) ground at 0.5 mm/min to a more densely fractured region ground at

1.0 mm/min. The cutting direction in this image is from top to bottom.

Force and Fracture Measurement The purpose of measuring the grinding forces is to verify assumptions made in the wheel parameter model and to gain insight into the ductile and brittle material removal processes. The workpiece was mounted on a three axis load cell, and cutting and thrust forces were measured. To identify the ductile-to-brittle transition, the surface of each region of the ground groove was evaluated with optical microscopy and computer image analysis software.

Soda Lime Glass Results

Results of areal fracture measurements on soda lime glass are shown in Figure 4. The transition from low percent fracture to high percent fracture indicates the location of d_c . Values of d_c for the 4-8 μm wheels are higher than for the 30-60 μm wheels which agrees qualitatively with the theoretical model.

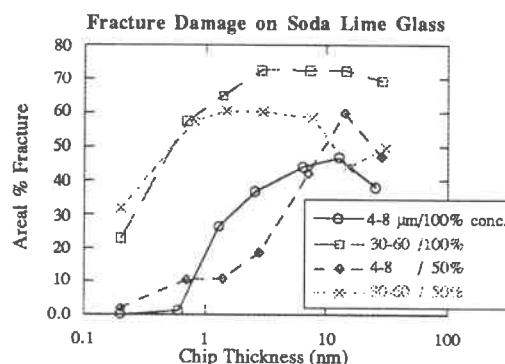


Figure 4: Fracture measurements of ground soda lime glass

Figure 5 compares the thrust force measurements for the four wheels. Since the depth of cut (and consequently the groove width) varied slightly for the four grooves, the thrust forces were normalized by dividing by the cross-sectional area of the groove. The wheel parameter model predicts that thrust force increases as grit size decreases or

concentration increases. Figure 5 corroborates this result. The small grit, high concentration wheel has the highest forces, and the large grit, low concentration wheel has the lowest forces. Tangential grinding forces were also measured and used to determine specific grinding energies. These measurements showed that for a given material removal rate (or given feedrate), the small grit wheels require higher energy than the large grit wheels.

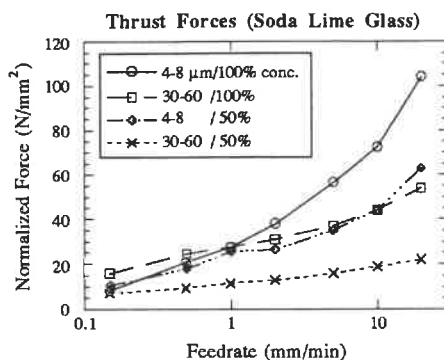


Figure 5: Grinding thrust force measurements

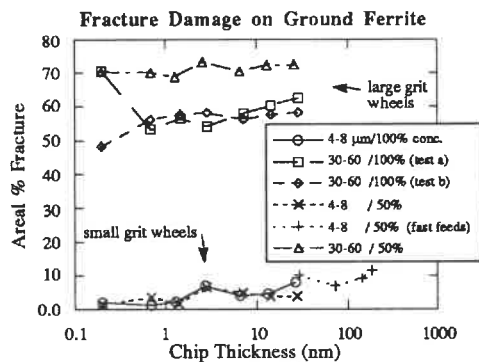


Figure 6: Areal fracture measurements for ground ferrite

Ni-Zn Ferrite Results

Results of areal fracture measurements on ground ferrite are summarized in Figure 6. The results show a strong dependence on the grinding wheel used and a minor dependence on chip thickness (or material removal rate). Unfortunately, this set of tests did not yield experimental values for d_c since no transition

from low fracture to high is apparent. These results seem to indicate that tests at smaller chip thicknesses for the large grit wheels (and vice versa for the small grit wheels) are necessary to find d_c .

CONCLUSIONS

The theoretical model developed above predicts that P_c (the critical indentation load based on material properties) and r_{grit} (grit size) have significant effects on the critical chip thickness for ductile-to-brittle transition. The experimental results presented showed qualitative agreement with the model but suggest that more experiments are required. Other researchers have conducted extensive analyses of the forces at the wheel/work interface in the grinding of *ductile* materials. The goal of most of this research was to predict form errors due to wheel deflection. The analysis presented above has adopted similar methods to achieve a different goal--prediction of the transition between the ductile and brittle regimes in the grinding of *brittle* materials.

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AN APPROACH TO IMPROVING CUTOFF ACCURACY IN OUTER-BLADE CUTOFF METHOD

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1. Introduction

The elastic deflection of blade under cutting has a great influence on the cutoff accuracy in outer-blade cutoff method. Such blade deflection is mainly caused by an asymmetric wear occurred on a cross-section of blade outer edge. If cutoff is carried out under a constant table speed in the configuration as in Fig.1 using a blade with an asymmetric wear, the blade deflection varies with the change of the contact area between blade and workpiece in its cutoff process. In result, the flatness of the cut-off surfaces becomes worse.

For the purpose of improving the flatness, this paper proposes a new cutoff method in which the table speed is controlled to keep the maximum blade deflection constant during one-pass cutoff process. We already discussed the method in the previous paper [1] in the case where the grinding force acting on blade side faces was ignored. But, it can not be ignored, when the thickness of diamond layer χ (Fig.2) of blade is large and the grinding performance of blade side faces is poor. Therefore, in this paper we discuss the new cutoff method considering the grinding force acting on blade side faces.

2. Theoretical analysis of new cutoff method

2.1 Control curve of table speed

Authors already developed a computer simulation method for the blade behavior under cutting. Using the method, the control curve of the table speed for keeping the maximum blade deflection constant can be calculated. In this calculation the z direction grinding force components f_{pz} and q_z , which denote the force acting per unit length along the contact arc AB and the force acting per unit area around the point P on blade side face respectively (Fig.2), are given by following equations.

$$f_{pz} = Cp \frac{\pi v}{2V} K_f \tan \bar{\gamma}_p \sin \theta \quad (1)$$

$$q_z = \frac{\pi}{2} C'p \frac{a}{Vr} u_s \tan \bar{\gamma}_s \quad (2)$$

where

- a : Radius of blade
- V : Wheel speed
- v : Table speed
- K_f : Asymmetric shape factor of blade, which expresses the amount of the asymmetric wear [2]

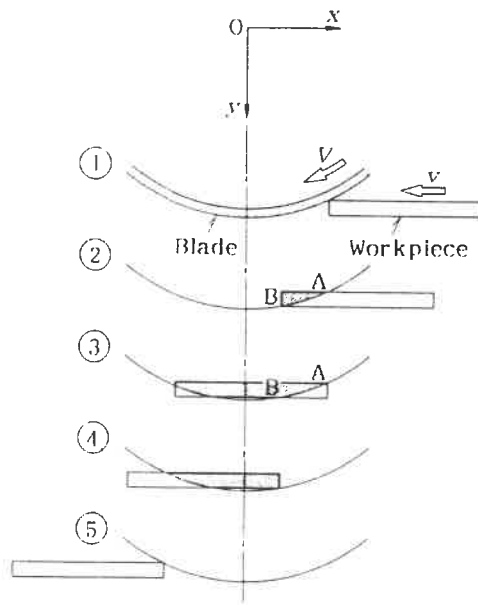


Fig.1 Cutoff process

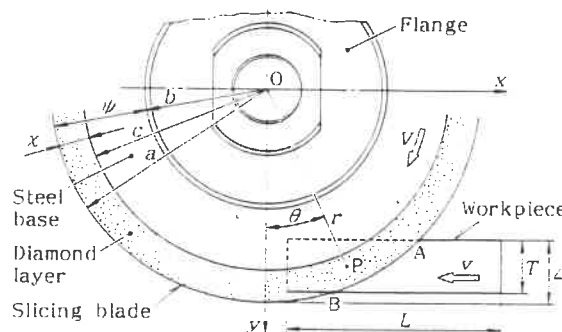


Fig.2 Arrangement of blade and workpiece

Table 1 Standard calculating conditions

Size of slicing blade	
Radius; a	62.5 mm
Thickness of diamond layer; x	3 mm
Width of diamond layer; L_d	0.5 mm
Width of steel base; t_b	0.4 mm
Size of workpiece	
Thickness; T	5 mm
Length; L	50 mm
Physical properties of blade materials	
Young's modulus of diamond layer; E_d	2.8×10^4 N/mm ²
Young's modulus of steel base; E_b	2.1×10^5 N/mm ²
Poisson's ratio of diamond layer; ν_d	0.25
Poisson's ratio of steel base; ν_b	0.3
Tangent of grain tip half angles;	
$\tan \bar{\gamma}_p$	30
$\tan \bar{\gamma}_s$	60, 100, 150, 200
Asymmetric shape factor of blade; K_f	0.087 mm
Cross-section shape factor of blade; κr	0.492 mm
Blade projection from flange; ψ	9 mm
Blade peripheral speed; V	1500 m/min
Depth of cut; Δ	6 mm
Cutting mode	Down cut
Specific grinding forces;	
C_p	2400 N/mm ²
C'_p	4500 N/mm ²
Friction factors between grain and work;	
μ_p	0.13
μ_s	0.28
Mesh size of blade side face and cut-off workpiece surface for calculation; Δs	0.5 mm

C_p, C'_p : Specific grinding forces in plowing on blade peripheral surface and on blade side face, respectively

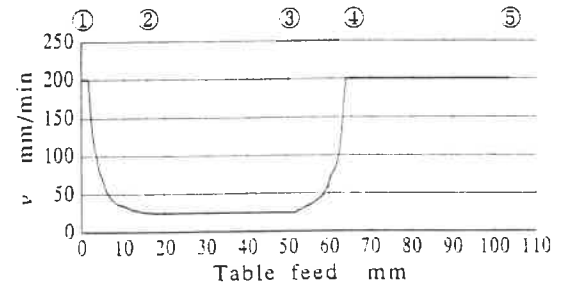
$\bar{\gamma}_p, \bar{\gamma}_s$: Average half tip angles of grains on blade peripheral surface and on blade side face, respectively

u_s : Volume of work removed per unit time by a unit area around the point P

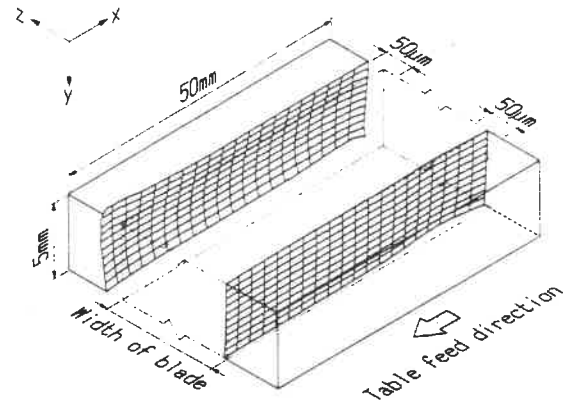
The control curve of v for keeping the maximum blade deflection $40\mu\text{m}$ and the cut-off surface forms were calculated under the conditions in Table 1, where 150 was selected for $\tan \bar{\gamma}_s$. The asymmetric shape factor K_f was given so as to deflect the blade in the positive direction of z axis, and the table speed was limited at 200 mm/min. The results are shown in Fig.3. The numbers ①–⑤ in Fig.3(a) correspond to those in Fig.1 respectively. Defining the feeding time of table from ① to ④ (Fig.1) as "cutting time", the cutting time of Fig.3 is 124 seconds. On the other hand, when cutoff is carried out in the same cutting time with Fig.3 by the conventional cutoff method in which the table speed is constant between ① and ④ (Fig.4(a)), the theoretical cut-off surface forms are shown as in Fig.4(b). Remarkable difference is recognized on the flatness between Fig.3(b) and Fig.4(b).

2.2 Theoretical estimation of new cutoff method

The proposed cutoff method was compared with the conventional cutoff method using the following estimate values.

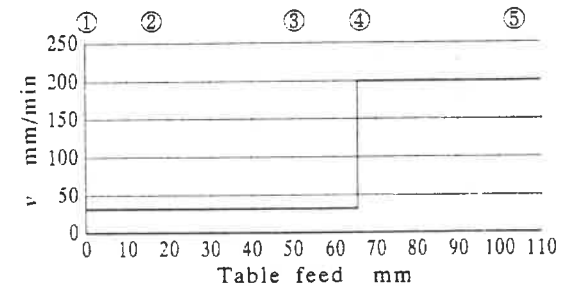


(a) Calculated control curve of v

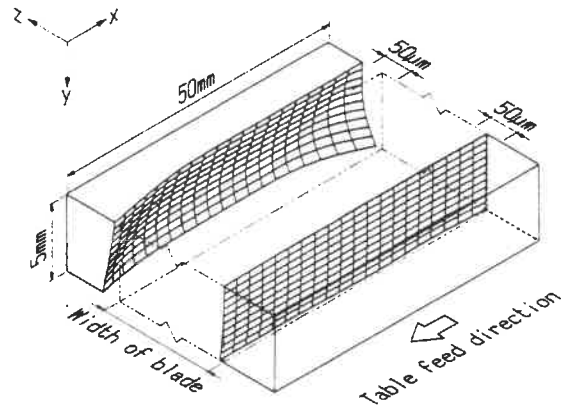


(b) Calculated cut-off surface forms

Fig.3 Analytical results of proposed cutoff method



(a) Variation curve of v



(b) Calculated cut-off surface forms

Fig.4 Analytical results on conventional cutoff method

D_B : Maximum blade deflection

W_{CM} : Maximum height of cut-off surface waviness
The distance of z direction between the peak and the valley in the estimated cut-off surface.

W_{CA} : Center plane average height of cut-off surface waviness

$$W_{CA} = \frac{1}{TL} \iint_{\Psi} |W_C(x,y)| dx dy$$

where Ψ is the region of estimated cut-off surface, and the function $W_C(x,y)$ is the deviation between estimated cut-off surface and its center plane which is perpendicular to z axis.

V_L : Curf loss

(Removed work volume) – $T \times L \times t_a$,
where T is the thickness of workpiece, and L is the length of workpiece, and t_a is the width of diamond layer of blade.

Superscript + (–) is written to the symbols W_{CM} and W_{CA} of the cut-off surface generated on the positive (negative) side of z axis.

The results under $\tan \bar{\gamma}_s = 150$ are shown in Fig.5, where the cutting times are equal in the proposed and conventional cutoff methods, and the cutting time 360, 207, 124, 93 seconds are required to keep the blade deflection 10, 20, 40, 60 μm respectively. In these figures, it is presented that estimate values except W_{CM-} and W_{CA-} become smaller in the proposed cutoff method than in the conventional cutoff method. Especially, the decreases of W_{CM+} and W_{CA+} are remarkable.

3. Experiment of new cutoff method

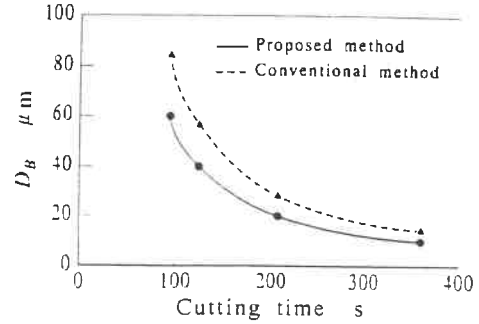
3.1 Experimental apparatus and methods

We made an experiment to test the efficiency of the proposed cutoff method.

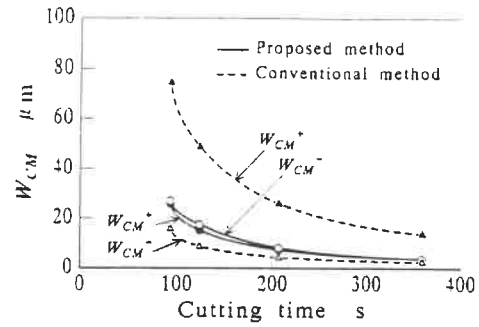
The grinding machine is improved in order to control the table speed v by computer according to the table feed distance.

In the experiment, the asymmetric wear of $K_f = 0.101 \text{ mm}$ was given to the blade outer edge in order to emphasis the blade deflection. Moreover, the blade side face was rubbed with a single point diamond dresser in order to deteriorate its grinding performance.

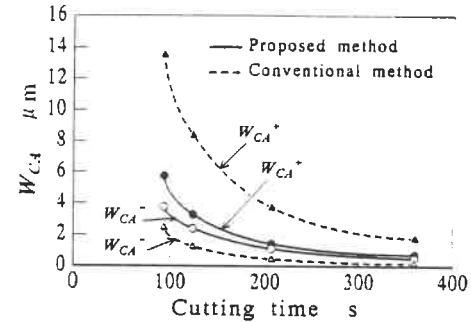
It is necessary to determinate the values $C_p \tan \bar{\gamma}_p$ and $C'_p \tan \bar{\gamma}_s$ for calculating the control curve of table speed v . For the determination of $C_p \tan \bar{\gamma}_p$, the grinding force component F_y , which is influenced very little by the grinding force acting on blade side face, was measured under 2 kinds of depth of cut D . Substituting the measured values of F_y in the theoretical equation of F_y [2], $C_p \tan \bar{\gamma}_p$ was determined 4700 N/mm^2 . Besides, $C'_p \tan \bar{\gamma}_s$ was determined by the maximum blade deflection D_B under cutting in the conventional cutoff method, which was measured from the obtained cut-off surface



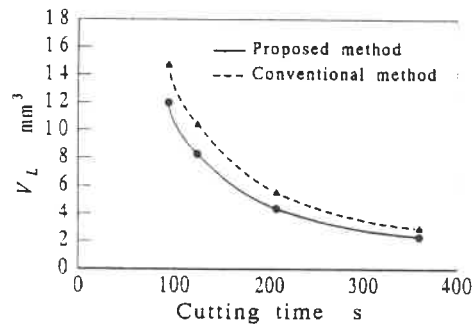
(a) Effect on D_B



(b) Effect on W_{CM}



(c) Effect on W_{CA}



(d) Effect on V_L

Fig.5 Analytical estimation of proposed cutoff method

form. In result we obtained $C'ptan\bar{\gamma}_s = 750000$ N/mm². The cutoff experiment was carried out using the control curve which might keep the maximum blade deflection 60 μ m. The proposed cutoff method was compared with the conventional cutoff method in the same way as we did in § 2. The grinding conditions are shown in Table.2.

Having the datum surface made of the guide way of the table feed and the vertical slide of the wheel spindle, the waviness of cut-off surfaces were measured on the grinding machine.

3.2 Experimental results

The cut-off surface forms obtained by the proposed cutoff method are shown in Fig.6(a). Besides, those obtained by the conventional cutoff method are shown in Fig.6(b). There is a great difference in the flatness between these cut-off surfaces generated on the positive side of the z axis. The estimate values, explained in § 2, of these cut-off surfaces are shown in Table 3. The value D_B in the proposed cutoff method became a little greater than 60 μ m which it should have been, but all estimate values except W_{CM} - decrease compared with the conventional cutoff method. These inclinations agree well with those of the analytical results shown in Fig.5.

4. Conclusion

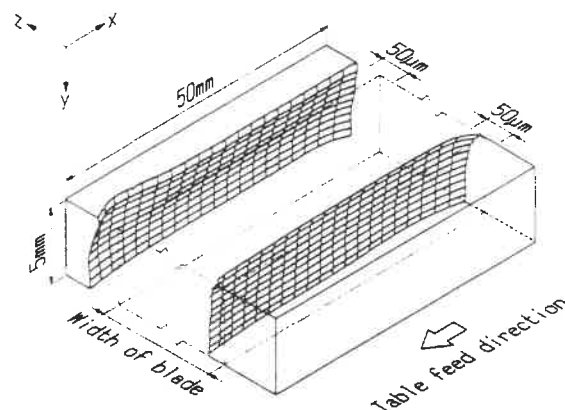
It is clarified from the analytical and experimental results that the proposed cutoff method, which control the table speed to keep the maximum blade deflection constant in one pass cutting process, is effective on improving the total flatness of the cut-off surfaces.

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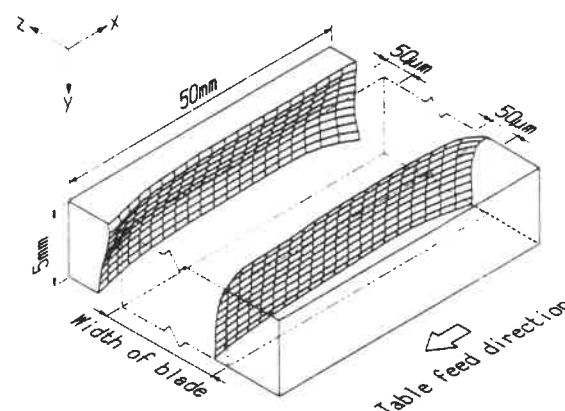
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Table 2 Grinding conditions

Slicing blade	SDC140V75BW6 (NORITAKE Co.,Inc.)
Size: $2a \times t_b \times (I.D.)$	125 x 0.5 x (38.1)
t_b	0.4 mm
x	3 mm
K_r	0.101 mm
κ_r	0.460 mm
ψ	9 mm
V	1500 m/min
Δ	6 mm
Cutting mode	Down cut
Workpiece	Ferrite
Size: $L \times T$	50 x 5
Grinding Fluid	Water soluble type



(a)By proposed method



(b)By conventional method

Fig.6 Cut-off surface forms obtained by experiments

Table 3 Estimation of experimental results

	Conventional method	Proposed method
D_B	76 μ m	67 μ m (-12 %)
W_{CM}	56 μ m	31 μ m (-45 %)
W_{CM}	53 μ m	53 μ m (0 %)
W_{CA}	9.2 μ m	6.6 μ m (-28 %)
W_{CA}	6.2 μ m	4.5 μ m (-27 %)
V_L	6.0 mm ³	5.2 mm ³ (-13 %)
Cutting time	78 s	

ULTRA-PRECISION GRINDING OF INORGANIC NONLINEAR CRYSTALS

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1. Introduction

A next generation laser fusion program is considered to use both high power laser diodes and third/fourth harmonic generation with nonlinear crystals in order to increase the energy efficiency, following the present program using the huge glass lasers and big KDP crystals. A potassium titanyl phosphate (KTP) is one of the best nonlinear material which is already used in second harmonic generation, and is usually finished by the conventional polishing technology which is a time consuming work. Therefore, it is very important to replace the polishing process with the ultra-precision grinding for realizing the next generation laser fusion program, as well as optical memories of high density.

This paper deals with the fracture/ductile mode grinding of inorganic nonlinear crystals and the possibility of making KTP single crystals into optical components only using grinding process.

2. Experiments

KTP crystals were grown by the top-seeded flux growth method [1] and cut into small samples of $9 \times 9 \times 2$ mm in dimensions at the phase matching direction. The ultra-precision surface grinder [2] having a glass-ceramic spindle of extremely low thermal expansion was used for perform the ductile mode grinding. The machine has a vertical spindle for a grinding wheel and a low-speed rotary table of 0.01-30rpm. Both spindles are suspended by hydrostatic oil bearings. The depth of cut can be numerically controlled at a unit of $0.1 \mu\text{m}$. Five kinds of cup-typed resinoid-bonded diamond wheels such as SD200-75-B, SD400-75-B, SD800-75-B, SD1000-75-B, SD1500-75-B were used in this study. The diamond wheel and workpieces were cooled with city water of 15l/min as a grinding fluid. The ground surfaces were observed with a Nomarski interference microscope of Nikon Corp. and measured in surface roughness with TOPO-3D of WYKO Corp. KTP crystals are hard ($H_v=566$) and brittle in nature.

3. Results and Discussion

Ground surfaces on Mn-Zn ferrite single crystals have been clearly classified into 3 modes, that is, fracture mode, ductile & fracture mode and ductile mode [3]. KTP samples were ground under the conditions of various feeds at the grinding speed $V_s=1200\text{m/min}$ and wheel depth of cut $a=1 \mu\text{m}$ using various kinds of diamond wheels. Figure 1 shows the relationship between the grinding modes and grinding conditions, and the relation was obtained after the observation of surface structures on ground surfaces with a Nomarski interference microscope. The ground KTP surfaces can be also classified into 3 modes. From Fig. 1, it is clarified that the grinding

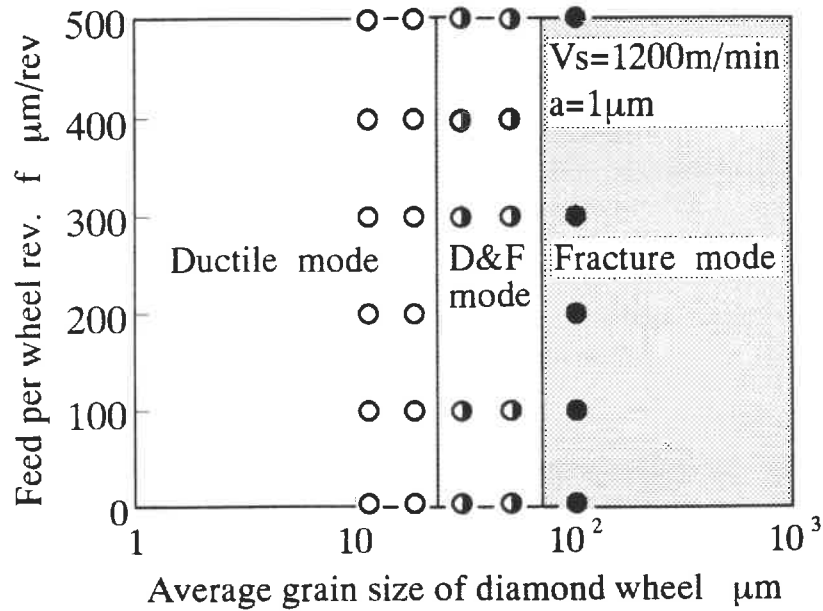


Fig. 1 The relationship between the grinding modes and grinding conditions.

mode strongly depends upon the grain size of diamond wheels and the feed per wheel revolution does not affect the grinding modes. The grain size ranges of SD1500-75-B and SD1000-75-B are $8\text{-}15\mu\text{m}$ and $12\text{-}25\mu\text{m}$, respectively. And both diamond wheels can get the ductile mode grinding as shown in Fig. 1, so it is concluded that KTP crystals can be ground in the ductile mode with the grinding wheels having diamond grains less than $25\mu\text{m}$ in grain size.

Figure 2 shows the relationship between the ground surface roughness and average grain size

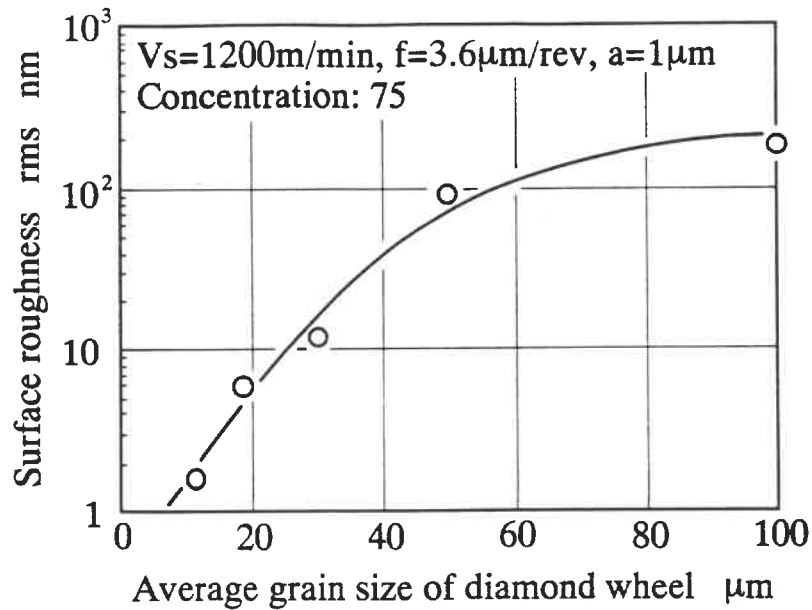


Fig. 2 The relationship between the ground surface roughness and average grain size of used diamond wheels.

of used diamond wheels. The surface roughness is strongly depends upon the grain size of the used diamond wheel, and it is clear that smoother surface may be obtained by using a diamond wheel having finer grains. An extremely large grain on the wheel working surface of a grinding wheel may worsen the surface extremely.

Figure 3 shows the relationship between the ground surface roughness and feed per wheel revolution at the ductile mode grinding. Smoother surface can be obtained by smaller feed per wheel revolution. It is also made clear that the ground surface roughness is not affected by the wheel depth of cut up to $10\mu\text{m}$ using SD1500-75-B wheel. Single crystals usually have the mechanical anisotropy, however, the ground surface roughness is the same at any grinding direction on KTP surfaces perpendicular to the phase matching direction in the same manner as Mn-Zn ferrite single crystals [3].

The ground surface roughness is affected by the grain size of the wheel and feed per wheel revolution, but not affected by the depth of cut and wheel peripheral speed in the ductile mode grinding. Therefore, it is important to increase the wheel depth of cut and wheel peripheral speed with decreasing the feed in order to machine the precision KTP optics efficiently.

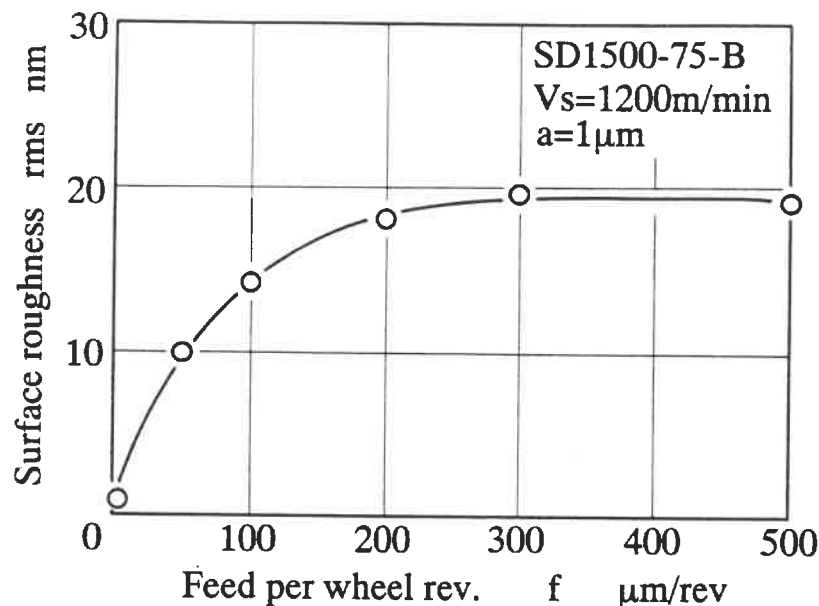


Fig. 3 The relationship between the ground surface roughness and feed per wheel revolution.

Figure 4 shows a surface topograph of ground KTP sample. The smoothest surface of 0.556nm rms , 4.31nm P-V better than optically-polished one can be obtained by one pass grinding with a SD1500-75-B wheel. The second harmonic generation can be obtained by the ground KTP crystal in a diode-pumped YAG laser system, so that the ultra-precision grinding can be considered to be one of the best process to get the optical surfaces on KTP nonlinear crystals.

It may be estimated that the ultra-precision grinding is able to finish various inorganic nonlinear crystals into optical surfaces besides the KTP crystals. This new technology is expected to be used as the final finishing process for inorganic nonlinear optics in the future.

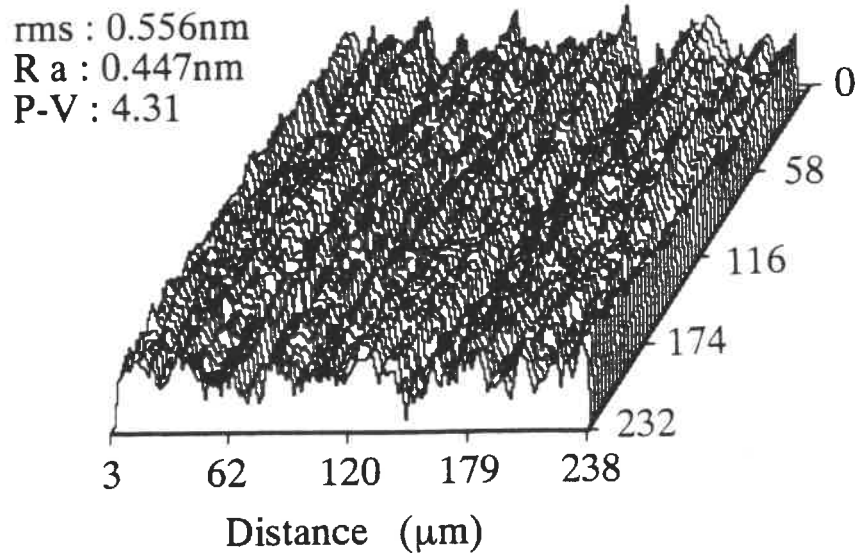


Fig. 4 The surface roughness of a KTP sample ground with a SD1500-75-B diamond wheel.

4. Conclusions

KTP inorganic nonlinear crystals have been ground by using the ultra-precision surface grinder and various resinoid-bonded diamond wheels to get optical surfaces. The following conclusions may be made on the results of this study.

1. Ground surface structures are strongly affected by the grain size of the used diamond wheels and the ductile mode grinding can be obtained by using a diamond wheel less than $25\mu\text{m}$ in grain size.
2. The ground surface roughness is a function of the feed per wheel revolution and average grain size of used diamond wheel, and is not affected by the depth of cut, wheel peripheral speed and grinding direction on the crystal in the ductile mode grinding.
3. Optical surfaces of 0.56nm rms in surface roughness can be obtained by the ultra-precision grinding with a SD1500-75-B wheel.

Acknowledgments

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VIRTUAL MODELING OF CONTOUR GRINDING

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INTRODUCTION

Contour grinding is an important process used to fabricate precision optical surfaces from brittle materials. In this process, a rotating grinding wheel is fed across a spinning part or raster-scanned across a stationary part to create a specified shape. Typically this process is used to generate the rough shape of the surface and is followed by a extensive series of loose abrasive grinding/polishing steps to form the finished optical surface. The objective is to produce a surface with the correct shape and a specular finish.

To speed up this process, ductile regime grinding has been proposed as an alternative to the loose abrasive grinding/polishing steps. The procedure uses a precision machine tool to control the motion of a fixed abrasive grinding wheel removing material in a manner that does not propagate cracks below the finished surface. As a result, the time required to produce an optical quality surface is greatly reduced.

To optimize the ductile grinding process, an understanding of the way the brittle material is removed and the limits of the process must be developed. A number of researchers in the US, Europe and Japan [1] are currently

working on this problem. The chief difficulty in developing this analysis is dealing with the complicated geometry of the chip. The chip shape results from a large number of overlapping interactions between the grinding wheel and the workpiece. Due to the complexity of this problem, there has been no concerted effort to develop a detailed model of this important material removal process.

This paper describes the effort under way at the PEC to develop a model of the precision grinding process. In addition the capability of a virtual reality environment will be utilized to visualize the influence of the wheel shape and operating conditions on the chip formation process.

GEOMETRY OF CONTOUR GRINDING

To develop a model for contour grinding, one significant assumption has been made: the features on the surface, and therefore the chip, are created once per revolution of the grinding wheel. This means that the grinding wheel can be visualized as a fly-cutter with one high spot that scoops out a region of the surface. While not necessarily a good assumption for all grinding geometries, it can be shown [2,3] to be valid for the small depths of cut and feed

rates used for ductile grinding of brittle materials.

Six variables are required to describe the geometry of contour grinding; wheel speed, part speed, crossfeed rate, wheel radius, nose radius, and part diameter. Of particular importance is the ratio of wheel speed to part speed, or phase. For small deviations of less than 0.1% in the wheel or part speed, dramatic changes occur in the surface and chip geometry [4].

Based on the assumptions described above, the interaction between the wheel and the part can be described as the removal of a scoop of material in the form of a chip, the shape of which is determined by the shape of the grinding wheel. The complicating factor is that the footprint of the chip is large with respect to the spacing between chips. As a result, the shape of any given chip is influenced by those around it with the number of interactions depending on the selected grinding conditions. To determine the shape of a single chip, the influence of all previous cuts overlapping that particular area of a part must be included.

Chip Geometry

To calculate the chip geometry, the intersection between each overlapping cut must be located and the shape calculated. This is analogous to watching the cutting operation take place through a window in the x and y plane in which all possible intersections can be observed. When viewed from the top, the intersection between two cuts of the toroidal grinding wheel appear as a straight line with an

elliptical outer boundary defined by the wheel geometry and the depth of cut. An example of a window displaying a two dimensional chip profile is shown in Figure 1. Once the intersections have been defined to create a theoretical grinding chip in the top surface plane, a third dimension is superimposed according to the shape of the grinding wheel and the location of the center of the wheel which produced each cut.

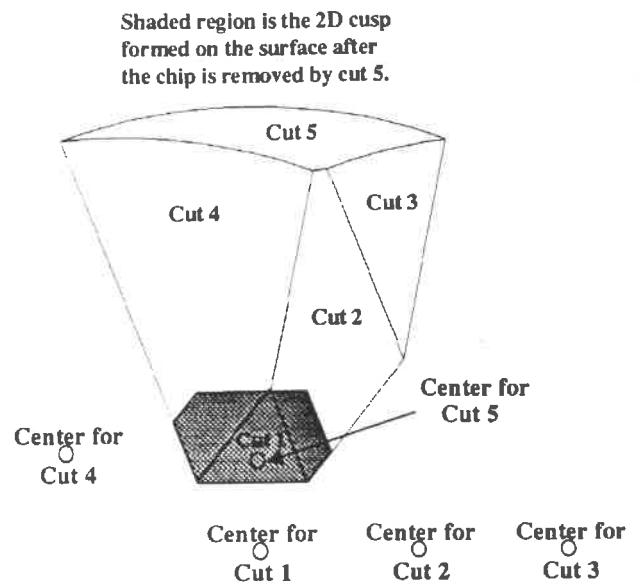


Figure 1: Two-dimensional chip profile displaying cuts created by intersecting revolutions of the grinding wheel.

Figure 1 shows the center locations of the grinding wheel producing the features on the top surface of the chip. In three-dimensions, the centers are several millimeters out of the page. Cut 5 is the final cut removing the chip and defining the shape of the bottom of the chip (not viewable from the top view). From Figure 1, a three-dimensional chip can be generated by superimposing a third dimension using the equation modeling the shape of the grinding wheel at each center location. Using

the x-y coordinates of the boundaries of the intersections and the center location of the grinding wheel during that cut, z coordinates are developed and a picture of the three-dimensional chip is produced. Figures 2 and 3 show the three dimensional chip developed from the two dimensional profile Figure 1 in front and two side views.

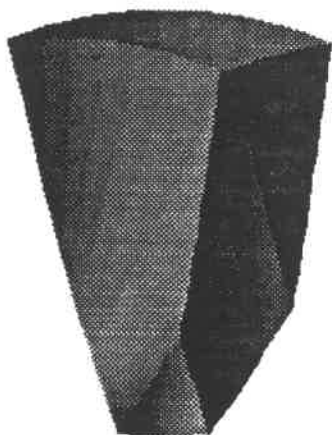


Figure 2: Shaded top view of three dimensional chip

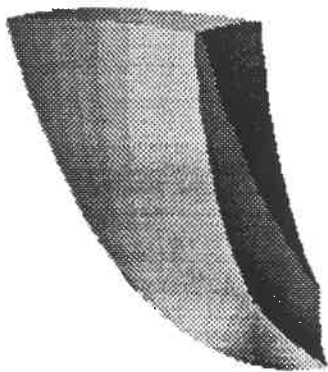


Figure 3: Shaded left side view of three dimensional chip

SIMULATING THE GRINDING PROCESS

To produce a simulation of the material removal process, the virtual grinding model proceeds in much the same manner as it does to create a single chip. The simulation continuously displays the formation of grinding chips, superimposing a third dimension after each revolution of the grinding wheel removes material from the surface. Intersections for each cut are calculated and displayed, and then the next is performed and displayed. The result is a virtual grinding environment in which the grinding operation appears to be removing material with each revolution of the grinding wheel. Parameters can then be changed to see the effect of each on the chip geometry and the surface finish as well as observe surface damage.

CAPABILITIES OF THE VIRTUAL GRINDING MODEL

The chip produced by the virtual grinding model can be used to calculate several important theoretical parameters. The final surface created is a direct result of the chip formation. The model calculates the shape of the surface and the statistical surface roughness parameters such as peak-to-valley and rms roughness. The thickness over any region of the chip can be calculated for theoretical comparisons with the subsurface damage generated during grinding experiments. The energy required to remove the chip can also be determined for use in predicting the grinding wheel lock-in speed proposed by Fawcett[3].

The theoretical parameters calculated by the virtual grinding model are important information to understanding the role of the geometry on the shape of the chip and the resulting surface features. However, the true benefits of this model come from the interactive virtual environment. This allows the researcher to examine the chip, surface, and simulated cutting operation simultaneously and interacting with the model by changing grinding conditions. When a parameter is changed, the effect can be seen immediately in the surface, chip, and surface. The transition from ductile to brittle material removal can also be displayed given a critical thickness value. This ability allows the process to be optimized by changing the conditions until the damage is confined to the chip and does not penetrate into the finished surface. The minimum energy required to remove each chip can also be determined which can lead to an optimum speed for the part and wheel. The virtual grinding model will be an invaluable tool in understanding the grinding process and the mechanisms that have the greatest effects on the surface finish and subsurface damage.

CONCLUSIONS

The virtual model for contour grinding has created an environment in which the grinding process can be visualized and the geometric mechanisms determining the surface finish and subsurface damage can be identified. The model allows a user to interact within a virtual environment by changing individual parameters to observe the effects on the surface, chip, and grinding process. From the model, any quantity relating to the geometry of the grinding process can be calculated such as

surface roughness, chip thickness, chip volumes, and grinding energy. These results can be used to develop optimal conditions at which to grind a material. The model is also versatile enough to allow other variables to be added such as vibrations and wheel runout. The ultimate goal is to predict the operating conditions best suited to finishing brittle materials to speed up the fabrication process while producing higher quality surfaces.

The modeling procedure developed for contour grinding is also applicable to other grinding geometries. A model for cup-wheel grinding to produce spherical surfaces is currently under development.

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The Influences of The Superficial and Structural Changes of Abraded Surface to The Toughening of $\text{ZrO}_2\text{-Y}_2\text{O}_3$ in Grinding Process

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It was made clear that tetragonal phase transforms to monoclinic and rhombohedral phases on grinding of ZrO_2 doped with 3 mol% Y_2O_3 . The bending strength σ_b of ground specimen is presented by $\sigma_0(1+\epsilon_{m+r}+\gamma)$, where σ_0 is the original bending strength of sintered matrix polished to surface roughness of 0.1 μm , and ϵ_{m+r} and γ are the each degree of influences by phase transformation and surface roughness. Furthermore, ϵ_{m+r} and γ were respectively defined by the following equations of $1.71(V_m-1.3)/100$ and $-0.55(R_{\text{max}}-0.1)/100$, where V_m is the volumetric percentages (%) of monoclinic phase and R_{max} is the roughness of abraded surface. The bending strength increased as much as 2 to 14 % by phase transformation and decreased as much as 2 to 5 % by an increment of surface roughness. The degrees of influences of ϵ_m and ϵ_r by both phase transformations were approximately 6 % and 5.8 %.

1. Introduction

Zirconia is regarded as the structural materials of machine element which provide the distinct properties of tenacity by the martensitic transformation from tetragonal phase(t-phase) to monoclinic phase(m-phase) or rhombohedral one(r-phase). Grinding process with diamond abrasive wheel has been one of the main techniques which machine zirconia effectively. However, it is still remained for an effective usage to make clear the influences of grinding process to the mechanical and structural changes of abraded surface of zirconia. The present paper mainly describes the numerical relationships between the superficial and structural changes of abraded surface and the bending strength. The possibilities of increase of mechanical properties induced by grinding process are discussed under the competitive comparison of surface roughness and a mass of phase transformation.

2. Experimental procedures

2.1 Originally sintered matrixes

A micro-powder of ZrO_2 doped with 3

mol% Y_2O_3 (HSY-3) was adopted in the present tests. Mixed powder with a binder was pressed in an uniaxial die under 10 MPa. After shrank in the hydroisostatic press, the original matrixes were obtained by sintering in air under the conditions of a heating duration of 180 min and a temperature of 1773 K. In four bending specimens($3\times4\times40$ mm) which were cut from a sheet of originally sintered matrix, two are original and two residuals are abraded by grinding process. A micro-structure of each specimen was investigated with the X-ray diffractometer(RU-200B). Eq.(1) gives the volumetric percentages of m-phase in a specimen.

$$V_m = \frac{I_m(\bar{1}11) + I_m(111)}{I_m(\bar{1}11) + I_t(111) + I_m(111)} \quad (1)$$

2.2 Grinding and oscilated wet-lapping tests

Table 1 shows the grinding conditions adopted in a grinding test with surface grinder(PSG-52DX). The roughness of abraded surface is scanned across the grinding scratches. Also, the sintered matrixes were lapped in the oscilator of SHINKO-SYNTRON. Loose abrasive grains of silicon carbide were used under the conditions of oscilated wet-lapping which are shown in Table 2.

Table 1 The conditions of grinding process

Wheel	SD325/400L100V SD140/170L100V SD50/60L100V
Speed	4.7, 9.4, 18.9 37.7 m/s
Table speed	0.17 m/s
Wheel depth of cut	5 μm
Cross feed	3 mm/stroke
Coolant	Yushiroken SC-18

Table 2 The conditions of oscilated wet-lapping

Grain	GC #800, #240, #100, #80, #36
Lap	Fe
Pressure	0.01 MPa
Workpiece speed	0.01 m/s

3. Results and discussions

3.1 Influences of grinding conditions to superficial and structural changes

Some factors of grinding conditions, such as grain size, wheel depth of cut, wheel speed and grinding direction etc., may influence to the superficial and structural characteristics of abraded surface. For example, Fig.1 shows the changes of m-phase contents and bending strength. According to the analysis with X-ray diffractometer, the greater height of peak of m-phase is with the increase of grain size, the smaller the peak of (111) of t-phase is. At a grain size of #50/60, the large cross cracks were observed on the abraded surface. Since such cracks relieve the transformation from t-phase to m-phase, then the height of peak of m-phase decreased.

It is natural that the roughness of abraded surface increases with the increment of grain size. However, at the extremely large sizes of #50/60, the volumetric percentages of m-phase decreased. Although the bending strength gradually increases with the increment of grain size, that contrarily decreases at a grain size of #50/60.

3.2 The degree of influence of the superficial and structural changes to bending strength

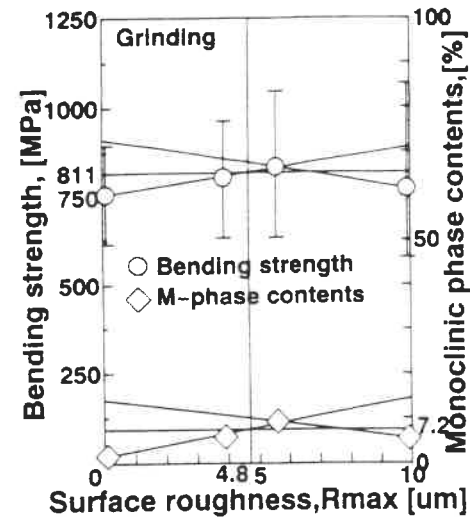


Fig.1 Changes of m-phase contents and bending strength

Table 3 The experimental and derived values of bending strength

	$V_m(\%)$	Rmax	Ex. σ_b
O.S.M.	1.3	0.1	755.6
Post-polish	7.2	0.1	832.0
Derived	7.2	4.8	811.0

The calculated bending strength $Ca.\sigma_b$ of ground specimens may be related with ϵ_{m+r} and γ which present respectively both degrees of influence by phase transformation and surface roughness, as shown in Eq.(2).

$$Ca.\sigma_b = \sigma_0 \{ 1 + \epsilon_{m+r} + \gamma \} \quad (2)$$

where, σ_0 is the bending strength of originally sintered matrixes.

The abraded surface ground with a wheel of SD140/170L100V was polished to the same degree of surface roughness as that of originally sintered matrixes. The experimental bending strength Ex. σ_b was 832 MPa. The derived values of bending strength at the same contents of m-phase will be searched from Fig.1 as 811 MPa. Then, Table 3 shows the experimental or derived values of bending strength. Comparing with the bending strength of both the originally sintered matrixes and the post-polished specimens after grinding, the differences of bending strength may be due to the contents of m-phase under the same

Table 4 Degree of influences by phase transformation and surface roughness

Wheel	Ex. σ_b MPa	Inc.(%) (Exp.)	ϵ_{m+r} %	γ %	Inc.(%) (Cal.)	Ca. σ_b (MPa)
O.S.M.	755.6	0.0	0.0	0.0	0.0	755.6
SD325/400	792.7	4.9	7.9	-2.1	5.8	799.2
SD140/170	833.5	10.3	13.7	-3.0	10.7	836.1
SD50/60	772.0	2.2	8.9	-5.4	3.5	782.1

roughness of abraded surface. Therefore, ϵ_{m+r} will be deduced as $1.71(V_m - 1.3)/100$, where V_m is the volumetric percentages of m-phase.

Since the differences between a bending strength of 832 MPa of the post-polished specimens after grinding and the derived values of 811 MPa is due to the increment of surface roughness under the same content of m-phase. γ will be deduced as $-0.55(R_{max} - 0.1)/100$.

Tables 4 shows the calculated results from Eq.(2) by putting the contents of m-phase and the roughness of surface into ϵ_{m+r} and γ . The toughening of ZrO_2 - Y_2O_3 increases with the positive effect of phase transformation and decreases with the negative effect of surface roughness. Furthermore, the increasing rate of calculated bending strength will coincide with that of the experimental bending strength of ground specimens against the originally sintered matrixes. It is clear that the influence of phase transformation is great to the toughening of ZrO_2 - Y_2O_3 in grinding process.

3.3 Separations of effect of m- and r-phase transformations by heat treatment

The (111) of r-phase is diffracted at the near angle with the (111) of t-phase in X-ray diffractometer. It is well known that r-phase is only formed on the machined surface¹⁾ and contrarily transforms to t-phase by heat treatment¹⁾.

First, after grinding with a wheel of SD140/170L100V, the abraded specimens were heated at every 100 K from 873 K to 1273 K with a heating rate of 300 K/hr, in order to decide a temperature of heat treatment. Figure 2 shows the changes of

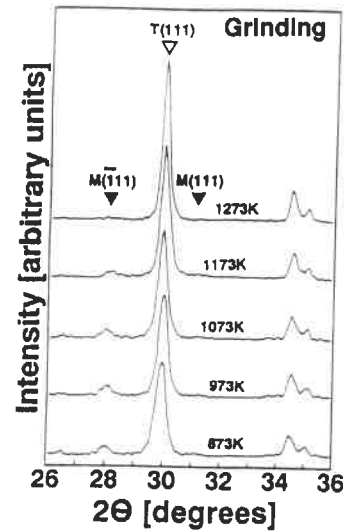


Fig.2 Changes of spectrum patterns of X-ray diffractometer

Table 5 Results of separation by heat treatment

	V_m (%)	ϵ	σ_b (MPa)
O.S.M.	1.3		755.6
Heat treatment	8.2	ϵ_m	801.0(Exp.)
SD140/170	8.2	ϵ_{m+r}	844.8(Cal.)

spectra obtained with X-ray diffractometer. The intensity of t-phase increases at higher temperature of heat-treatment above 873 K with the sharpened shape of spectrum. In other words, the contrary transformation proceeds from r-phase to t-phase by thermal energy. The intensity of m-phase increased at a lower temperature below 1273 K. The martensitic transformation from t-phase to m-phase will occur about 1273 K and m-phase is stable below 1273 K. Therefore, it was decided that 1073 K is the moderate temperature of heat treatment at which the peak of spectrum of t-phase is sharp, r-phase loses and m-phase does not transform to other phases. Table 5 shows how the existence of r-phase influences to the bending strength at V_m of 8.2 % and surface roughness of 0.1 μm . The experimental bending strength Ex. σ_b was compared with the calculated bending strength Ca. σ_b which was obtained by putting V_m of 8.2 % and R_{max} of 0.1 μm in both equations

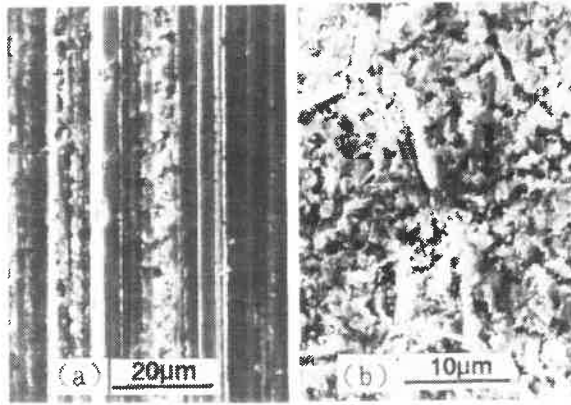


Fig.3 The abraded surface by grinding process and oscillated wet-lapping

of ϵ_{m+r} and γ . ϵ_{m+r} was 10.8 % which consists of ϵ_m of 6 % and ϵ_r of 5.8%. Phase transformations from t-phase to m- and r-phases contributes each half of full on the toughening of $ZrO_2-Y_2O_3$ to the originally sintered matrixes.

3.4 Relationships between fracture mode and phase transformation

As shown in Fig.3(a), the abraded surface is formed by both of plastic deformation and brittle fracture in grinding process. On the other hand, Fig.3(b) shows the surface of $ZrO_2-Y_2O_3$ which was lapped in wet. The brittle fracture of abraded surface is observed with the severe unevenness without plastic deformation. ϵ_{m+r} and γ in the oscillated wet-lapping were obtained by the same treatment as that of grinding process. Then, $Ca.\sigma_b = \sigma_0 \{1 + 1.08(V_m - 1.3)/100 - 2.56(R_{max} - 0.1)/100\}$.

ϵ_{m+r} and γ were also calculated from the experimental values of V_m and R_{max} . Table 6 shows the calculated values which provide the positive effect of phase transformation and the negative effect of surface roughness to the toughening of $ZrO_2-Y_2O_3$. The experimental increasing rates of strength coincide with the calculated values by the above derived equation. Here, the influences to the toughening by both of m- and r-phases were investigated by heat treatment under the same conditions as that of grinding process. In the

Table 6 Degree of influences by phase transformation and surface roughness

Grain	Ex. σ_b MPa	Inc.(%) (Exp.)	ϵ_{m+r} %	γ %	Inc.(%) (Cal.)	Ca. σ_b MPa
GC#800	782.1	3.5	10.4	-6.9	3.5	781.7
GC#240	851.9	12.7	25.9	-12.8	13.1	854.7
GC#100	852.8	12.1	32.7	-18.7	14.0	861.7
GC#80	824.7	9.1	31.4	-20.0	11.4	842.2
GC#36	899.0	19.0	32.3	-13.1	19.2	900.9

oscillated wet-lapping, ϵ_{m+r} was 12.9 %, ϵ_m was 10.1% and ϵ_r was 2.8 %. This means that the degree of influence of transformation from t-phase to m-phase is approximately three-fourth. In the oscillated wet-lapping, the transformation is much from t-phase to m-phase and is little to r-phase. Therefore, the brittle fracture relates with the martensitic transformation from t-phase to m-phase, as well as the plastic deformation relates with the transformation from t-phase to r-phase.

4. Conclusions

The following conclusions are obtained through the investigation about the influences of superficial and structural changes of abraded surface to the toughening of $ZrO_2-Y_2O_3$ in grinding process.

- (1) The contents of martensitic transformation increases with the increment of grain size and wheel speed, but are not influenced by wheel depth of cut.
- (2) Phase transformation from tetragonal phase to monoclinic phase and rhombohedral phase contributes each half of full on the toughening of $ZrO_2-Y_2O_3$ to the originally sintered matrixes.
- (3) Tetragonal phase transforms to monoclinic phase by the brittle fracture and to rhombohedral phase by the plastic deformation in grinding process.

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AN APPLICATION OF COGNITIVE SCIENCE TO MAKING THE KNOWLEDGE BASE FOR GRINDING

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1. Introduction

The knowledge acquisition is the most difficult problem in making the knowledge base of the expert system. To solve this problem, we change the stand point from the knowledge acquisition of the expert, which is the usual approach, into the knowledge acquisition by the expert. Namely, we investigate the learning and problem-solving processes by the expert. We assume that in these both processes, the text written in natural language plays a very important role. We read the text and acquire the knowledge from it. We produce the text as a result of the problem-solving. Thus, the knowledge acquisition problem is finally change into the well known problem in cognitive science, namely, how to read the text and how to write the text. Also, this problem is intimately related with text linguistics (1) and semantics (2).

In this study, we investigate some texts of machining (cutting and grinding). The first problem is to find the general structure of the words used in the text. The second problem is to find the relations between the condition sets

and the phenomenon sets as the result of realizing the text written in natural language. The final problem is how to interpret the text from the stand point of the hierarchical modeling. The hierarchical modeling contains the mathematical model, the physical model, the engineering model and the empirical model.

From above results, we can make the highly structured knowledge base from the text which is written in natural language.

2. General Structure of Words

We read the text by using the word system which has been acquired already in the past. If the acquired word system is domain general as large as possible, we can more easily understand the various domain special texts (2). In this research, we try to construct the domain general word system. Most texts are constructed in the general form. That is, it has the hierarchical table of contents and the index. From some various cutting and grinding texts, we derive the general structure of words. Grinding or cutting is a kind of human activity, therefore the general structure of words for

grinding or cutting are constructed as follows.

The six elements of "act" are the purpose, object, means, condition, procedure and estimation shown in Fig.1. We also propose that act, entity, condition, procedure, phenomenon and relation are as fundamental words in order to construct the category-like word system giving consideration to the specialty (for example, grinding or cutting). Among these six word types, the entity and the activity are more fundamental than others. Fig.2 shows word structures of entity and relation, for example.

3. Relations

The text usually deals the special theme with

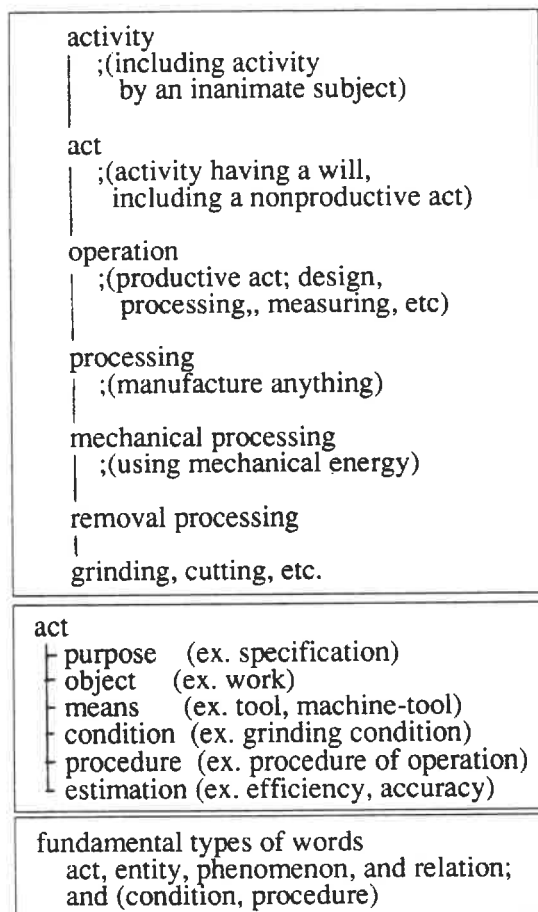


Fig.1 the category sequence of activity, "act" structure and the fundamental wordssystem.

using natural language. It contains various kinds of sentences, for example, simple sentencea, complex sentences and compound sentences. And also, various kinds of conjunctions are used in the text. We consider the text as the set of various kinds of relations among fundamental simple sentences. We simplify the text according to the following procedure.

1. Fundamental simple sentences and three type of conjunctions; "and", "or" and "if_then" ;The most important points at this stage are that we forbid to use the conjunction of "but" or "however" which appears very frequently in the natural language text

2. S_expression; namely, ((the subject part of a sentence)(the primary predicate part)(the rest of a sentence)), and so on.

;At this stage, the structure of the simple sentence is regarded as the approximation for describing the object.

3. W_functional_notation; (Relation (the subject part of a sentence) (the subject part of another sentence))

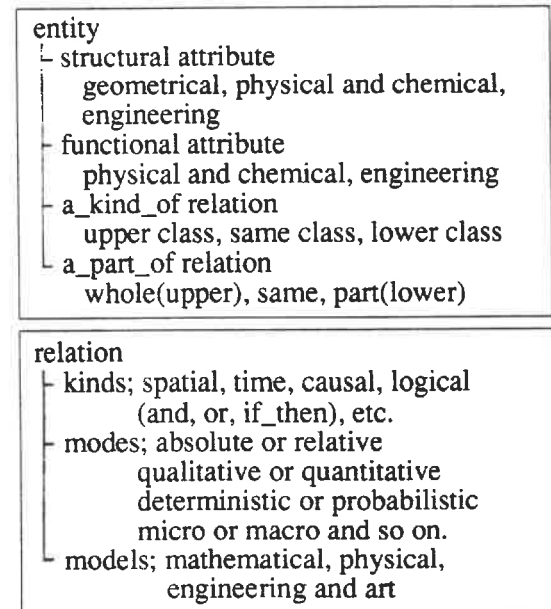


Fig.2 Word structures of entity and relation

Thus, we obtain the final approximated structure for each sub-texts. The text learning process could be regarded as a kind of the chunking process of the complex relations describing the theme. The text constructing process also could be interpreted as the process of specification on the theme.

4. Hierarchical Modeling

The sentences in the text is pursued above as the sets of the relations which correspond to the formal aspect of the text. On the other hand, in order to describe the contents of the text, we already apply two concepts about the structure of the objects (in this case, words), namely, a_kind_of structure and a_part_of structure. Further, we propose the hierarchical model as the third structure of the relations used in the text. Considering the knowledge about principles and laws which regulate behaviors of the target system is very important in the construction of the knowledge base(3). The natural language is very strong and flexible as the means of expression on the subject at the empirical level. But, even at the empirical level and still more at the engineering level, we consciously (or unconsciously) use (or obey) the various model, for example, mathematical model, physical model, engineering model and empirical model(or art model). Therefore, in order

to construct the knowledge base, we should explicitly and consciously adopt the hierarchical model to the text transformation. Fig.3 shows the schematic three dimensional space for the structured knowledge base.

5. Typical Example of Structured Knowledge Base

The original sub-text, describing the various chip formation observed in metal cutting, contains 32 sentences in Japanese. We transformed them into 130 fundamental simple sentences. Statistics of vocabulary are as follows (parts of speech);

	number of use,	number of kind
noun	402 times	120 kinds
verb	42	16
adjective	47	17
adverb	32	14

From these result, noun is the most important element in this sub-text. So, if we succeed in making the structured word system, it is easy

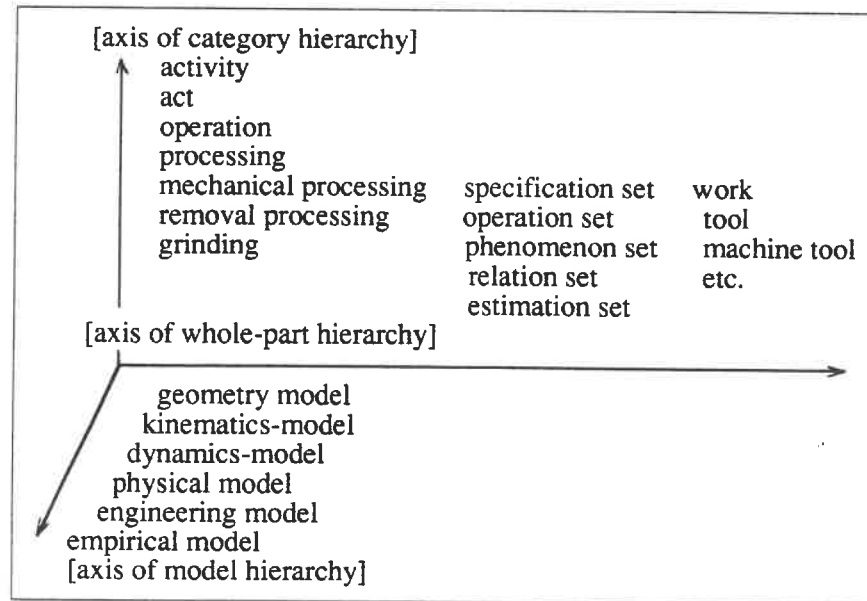


Fig.3 Schematic of Three dimensional space for constructing knowledge base

to construct the structured knowledge base from the text in natural language. Fig.4 schematically shows the structured knowledge base derived from this sub-text according to the word system which is explained already. The structures of subject parts in each fundamental simple sentences approximately coincides with the word system. For example, some typical subjects parts are (tool rake-angle), (work material), (cutting-condition depth-of-cut), and so on.

If once we make the structured knowledge base for machining in the three dimensional space, we can use the inherited knowledges from the texts which belong to the upper field and differentially construct the knowledge base. And it is possible to infer qualitatively with using these knowledge base. We can also learn or reason by analogy with using the

knowledge bases which are constructed in the vicinity of the objective field. And further, the knowledge which is not described precisely or omitted in the text could be pursued systematically using the three dimensional knowledge base space.

6. Conclusion

We have already had huge volumes of the texts for various specialties in the many different fields. Therefore, if we succeed to computerize the knowledge which are contained in these text, we can obtain unbelievable quantities of benefits concerning not only the knowledge acquisition but the text learning process.

For this purpose, we proposed the word system, the procedure of transformation of the text described in natural language into the struc-

tured relations in order to computerize the text knowledge, and finally, the three dimensionally structured knowledge base system.

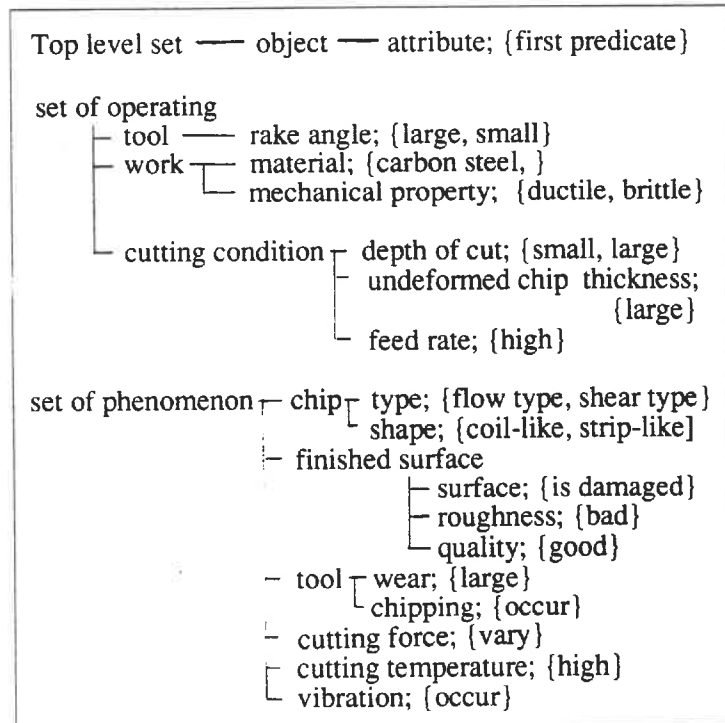


Fig.4 Schematic representation of the structured knowledge base derived from this sub-text according to the word system

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ELID DRESSING OF GRINDING WHEELS

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Metal bonded grinding wheels are typically used for their ability to take heavy loads without the hazard of wheel failure, and for the better grit strengths. Truing and dressing these wheels, however, is difficult. Research is being conducted in the use of electric discharge or EDM dressing, electrochemical (ECM) dressing, and others. Each has its merits, but we have chosen to research electrolytic dressing, nicknamed ELID dressing, which causes the steel or iron bond to form oxide products on the surface of a wheel. These products inhibit the electrical action, halting or slowing the dressing process when the wheel is not being used. This mechanism is equally applicable to very fine grinding where typical wheels load up with contaminants or break down quickly, making their use difficult. Our application and my research are in this fine grinding of finish quality surfaces in hard mold materials and aspherical surface grinding of glass lenses using ELID.

ELID requires a cast iron or steel bonded wheel, a pulsed DC power supply, and coolant that can both conduct electricity and promote the proper chemical reactions at the grinding face. The current is given a path between the positively charged wheel through the coolant in a gap, to the negatively charged electrode. ELID has been demonstrated on oscillating surface grinding machines, OD grinders, and Blanchard style surface grinders, among others. The only requirement for a particular machine or application is that there must be access to a portion ($1/4 - 1/6$) of the grinding wheel grit surface for the electrode.¹ The only sources for wheels, coolant, and power supplies that are currently proven and available are in Japan, where ELID is being used or tested in many companies, mostly for rough grinding operations, but also in making precision flats, forms, and aspherical lenses.

The wheel goes through two phases of dressing using ELID. Phase one is a pre-dressing, which removes the bond matrix for $1/3$ to $1/2$ of a grit particle size, exposing the grit for use. This phase ends automatically in the creation of an oxide layer which inhibits the electricity flow. The second phase occurs during grinding. As the oxide layer is removed by the grinding action, the electrical current is restored to the portion of the wheel in use, causing further oxide creation in this zone and a fine truing of any remaining runout from the wheel. The process continues to dress the bond in the regions where the grit is used, and wheel loading can be eliminated. Halting the grinding action results in the creation of a new oxide layer and the dressing is stopped automatically. This inherent ability results in longer wheel life and a larger grinding ratio. In grinding aspherical parts with the wheel OD, changes in the wheel diameter during the grind introduce errors in the surface that is produced. By minimizing the wheel wear, these introduced errors are also minimized.

Wheels with brass bonds do not show the nonlinear dressing current effect associated with iron based wheels. In this case the bond is removed whether there is grinding action or not. I would prefer not to call this ELID, to separate the two processes. Perhaps electrochemical dressing could be used to refer to this constant current removal process.

A standard RTH diamond turning lathe can be fitted with a grinding attachment for the generation of precision ground aspheres or other rotationally symmetric surfaces in components that are not diamond machinable. This attachment has always used a resin bonded diamond or CBN wheel, because we had no way to properly true and dress metal bonded wheels, which resulted in chatter in the surfaces. Compared with resin bonded wheels, ELID dressing allows us to achieve the same figure error with better surface finish, while eliminating redressing operations and improving throughput.

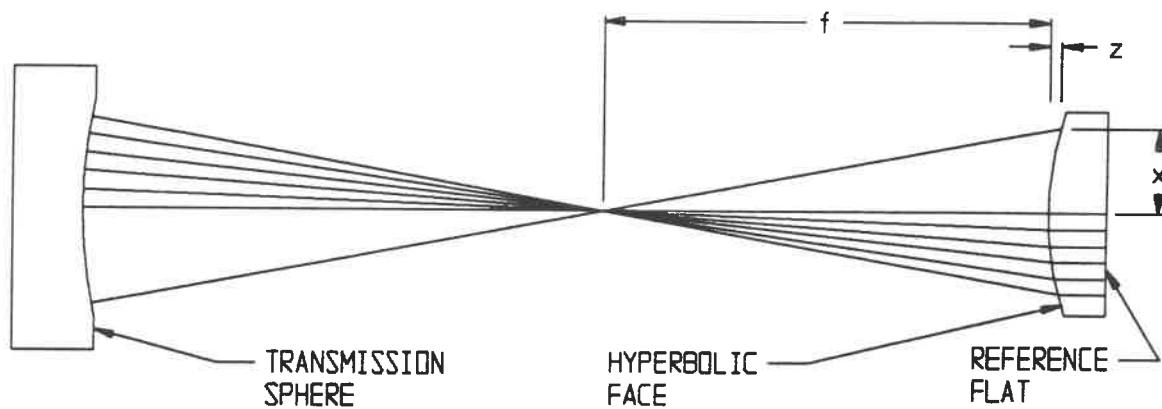
In ELID, the oxide created is assisted by the electricity that flows in one direction, on the positive side of the electrode gap. An AC voltage placed across the gap will cause the oxide to form, but with a very thin depth, which is not suitable for both exposing the grits and forming an insulating layer. A constant voltage DC source also creates a thin oxide layer and does not penetrate enough. As the frequency of pulsing the DC voltage is increased, the depth of penetration increases. A pulse rate of 4 microseconds on and 4 microseconds off is typical in ELID dressing. A unit designed for EDM use may prove suitable for ELID dressing, although the current draw will be less than typical EDM currents. Some units have been designed specifically for ELID, but are available only from Japanese companies at present.

The coolant is probably the most crucial element of the ELID process. It must have the ability to carry electricity, which is not normally a feature of water based coolants. It must also have the right ionic additives such as dissolved chlorine to promote the oxidation of the bond matrix. However, the presence of calcium can have a detrimental effect, causing undesired buildup on the wheel. Because the water used is usually supplied from the nearest faucet, this represents an undesired variable in that the water quality can vary by season and test results are valid for only the sample in question. Water testing is recommended initially, but in usage it still may be best to change the coolant more often rather than trying to monitor and adjust the multiple properties (PH, hardness, conductivity, sulfate, chlorine, etc.) required.

The configuration of wheel and electrode is machine dependent, but for our machine I made the electrode such that the wheel could be changed without disturbing it. In this manner a roughing wheel could be used for the generation of the proper shape and a finishing wheel used for obtaining the surface finish without requiring complete disassembly to change the wheels. Adapters were designed for the wheel and electrode to insulate both from the machine chassis. I have seen machines working with only the negative (electrode) side insulated, but feel that the corrosion products generated on the wheel could also be encouraged on the machine chassis in such a setup. Insulating the wheel adapter required a rigid insulator, and a machinable ceramic was used. This also required the use of a brass brush ring on the insulated portion of the wheel adapter to carry the current to the wheel. The last requirement is to fill the electrode gap with coolant in order to carry the current flow. This can be difficult at high wheel periphery speeds and a special coolant manifold was made to flood an enclosed cavity surrounding the gap in order to hold coolant in the gap.

This arrangement was designed to be easily retrofitted to our ASG-2500 and Nanoform 300 machines. The hardware mounts into existing bolt holes in the machine and consists of two assemblies. One is the grinding wheel adapter, which contains balancing holes, a brass brush ring, captive mounting screws, and is removable from the spindle without disassembly of the adapter. The other is mounted next to the spindle and contains the coolant manifold, the brushes that carry current to the spindle, and the copper electrode in the shape of the wheel OD. The electrode is adjustable to obtain the proper clearance with the wheel OD and in height to match the wheel height. It is 45 mm long to surround about 1/5 of the 75 mm diameter wheel's circumference. There are two wires which carry the ELID current from the power supply outside the machine enclosure to the stationary assembly.

After early testing with flats and spheres, a hyperboloid of revolution in a glass substrate will be ground. By choosing the conic parameters based on the glass index of refraction, the part can be tested on an interferometer with a spherical wavefront. The hyperbola parameters are chosen so that the wavefront is converted from spherical to planar upon refraction through the glass interface. The wavefront then reflects off a reference flat on the back side and returns on the same path for interferometric analysis of the errors in the asphere. ²

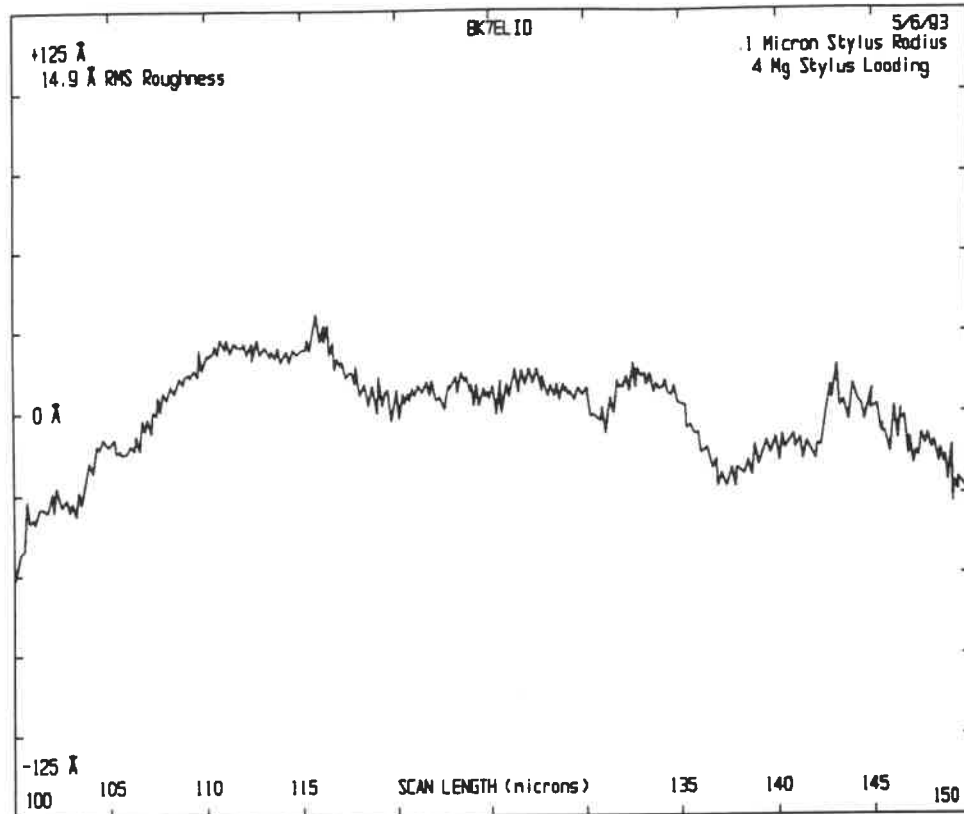


The hyperbola is defined by requiring the axial optical path length of $zn + f$ be equal to the path of the off axis rays as shown:

$$zn + f = \sqrt{(f+z)^2 + x^2}$$

where n is the index of refraction of the glass part. This relationship can be derived to one where the vertex radius of the part is $r = f * (n - 1)$ and the conic constant $k = -n^2$, or

$$z = \frac{r - \sqrt{r^2 - (k+1)x^2}}{(k+1)} = \frac{f(n-1) - \sqrt{f^2(n-1)^2 - x^2(1-n^2)}}{(1-n^2)}$$



This figure shows 14.9 Angstrom RMS surface roughness taken on a Rank Taylor Hobson Talystep instrument. The part is a BK-7 glass lens with an OD of 50 mm and a 110 mm radius spherical surface ground using ELID. It was ground on an RTH ASG-2500 machine with an 8000 mesh (1.8 micron diamond grit) cast iron bond wheel made by the Fuji Die Company. The electrode clearance to the wheel was 0.1 mm. The power supply was a Sintoblator EDM unit modified to output 60 volts and set at 2 microseconds on time and 5 microseconds off time. The predressing current was a maximum of 1.4 amps and decreased to an ELID dressing current of 0.2 amps. The coolant was Noritake AFG-M at 50:1 dilution with spring water. The spring water properties were: 8.1 PH; 57 ppm hardness; 130 μ S/cm conductivity; 8.1 ppm chlorine; and 17 ppm sulfates.

By utilizing the hydrostatic slide ways and the air bearing work spindle of our standard Nanoform 300 machine, ongoing efforts are to use ELID in the generation of optical aspheres which do not require polishing after grinding.

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BRITTLE/DUCTILE SURFACE GRINDING

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1. Introduction

Ceramic materials are becoming more popular in industry and production, replacing traditional metals. The exceptional properties of these materials make them highly suitable for more and more new applications. The main drawbacks with these materials is their poor machineability due to their combined hardness and brittleness. The mechanism of chip removal for ceramics is different than in the case of metals, which often behave as a ductile material.

The chip removal is considered brittle if the dominant mechanism is fracture and cracking [3] and ductile if the dominant mechanism is plastic deformation. In each case there is a small percentage of plastic deformation, respectively fracture.

According to Subramanian [4], the grinding process can be divided into brittle and ductile regimes. Subramanian also considers that in most cases the grinding of ceramics is in a combination of both brittle and ductile regimes and because of this there are some limitations regarding the quality and integrity of the ground surfaces, some of which are the focus of this paper.

2. Experimental

A surface grinder, type Rubin VA II (ELB Schliff) was equipped with instrumentation for measuring the grinding forces, vibration, velocities, diamond wear and consumed energy during the wet grinding of Al_2O_3 ceramic and K20 metal carbide.

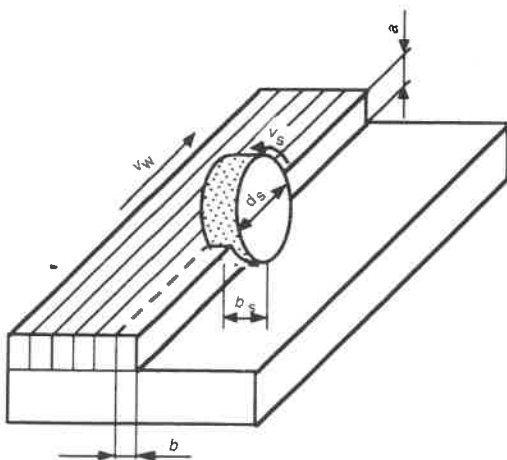


Fig. 1

Figure 1 presents a schematic of the surface grinding set up for the experiment and the main parameters measured.

3. Test Results

In Figs. 2 and 3 the results of grinding ceramic (2) and metal carbide (3) are presented for fine grinding. International CIRP notations and relationships are used for the technological parameters [1, 2]. For both materials diamond wheels with a resin bond were used. On the same graph, in order to visually optimize these axes, are

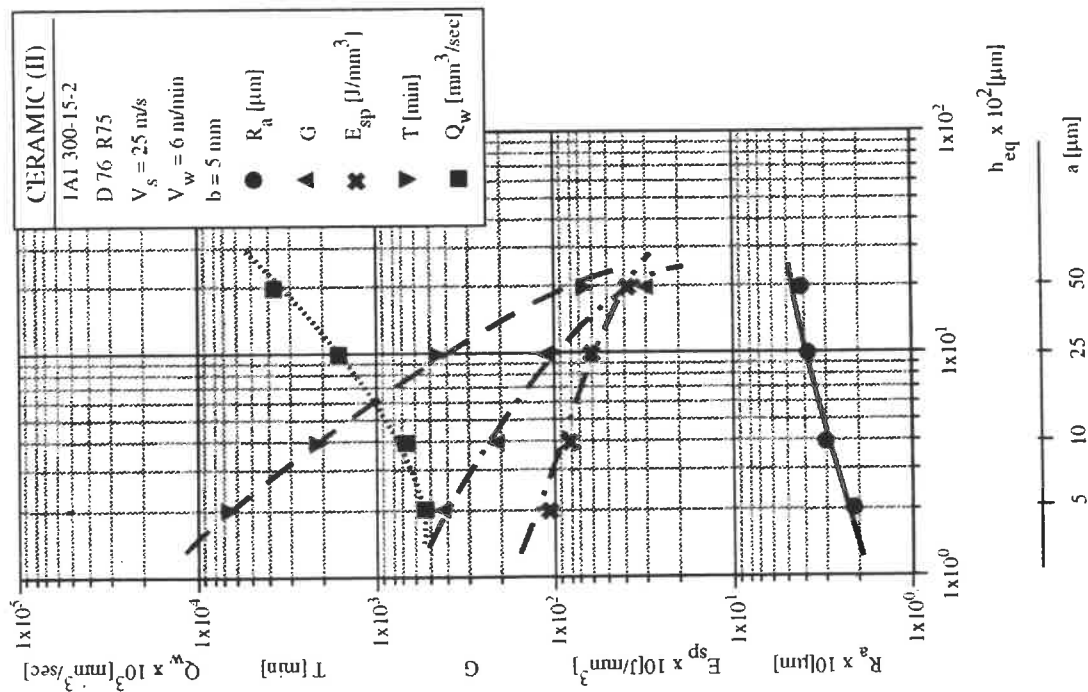


Fig. 2

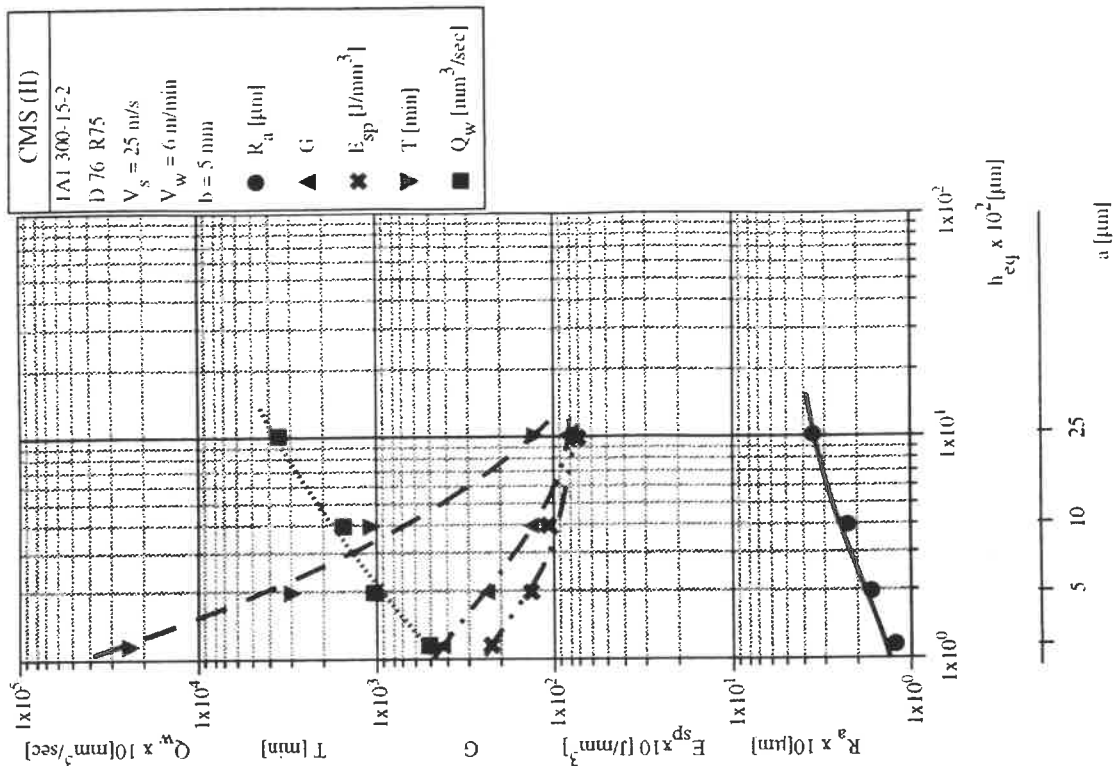


Fig. 3

plotted the main technological parameters: R_a , G (grinding ratio), E_{sp} , Q and T in log-log coordinates. Using a mathematical method, a polytropic relationship has been established for R_a and G , using h_{eq} (equivalent chip thickness) as an independent variable.

Ceramic

$$R_a = 0.375 \cdot h_{eq}^{0.029}$$

$$G = 97.791 \cdot h_{eq}^{-0.101}$$

Metal carbide

$$R_a = 0.2319 \cdot h_{eq}^{0.0226}$$

$$G = 116.38 \cdot h_{eq}^{-0.0395}$$

These relationships enable the influences of h_{eq} (which is a function of a , v_s , v_w) to be seen upon the main parameters of the grinding process.

Directly from the graphs, we can see that there is an optimum zone for both materials. For the ceramic the optimum is around $h_{eq}=0.065 \mu m$ and for metal carbide, the optimum is around $h_{eq}=0.03 \mu m$. For these values, we can obtain a good surface quality (relatively) at the same time as very good wheel life, efficient stock removal rate and low specific energy.

For both materials analyzed, SEM photography shows that they are both in a ductile grinding regime, but in the case of ceramic, the percent of fracture is higher than in the case of metal carbide (Fig.4).

This observation is also evident from Fig 5 where chips from grinding are presented for metal carbide and ceramic. We can observe the plastic deformation flow in the case of metal carbide and a greater proportion of fracture in the case of ceramics.

Conclusions

Utilization of equivalent chip thickness h_{eq} to present the main technological parameters, in double logarithmic coordinates, permits the optimization of the grinding process by computer for machine control.

Taking into account the technological and economical aspects for optimum grinding of ceramic in a predominantly ductile regime, a diamond resin bond wheel has been shown to be suitable.

This comparative study of fine grinding ceramic and metal carbide helps to understand the stock removal mechanism in the case of ductile and brittle materials. Pure brittle or ductile grinding does not occur and the quality of surfaces is dependent on the percent of fracture removal mechanism to ductile removal.

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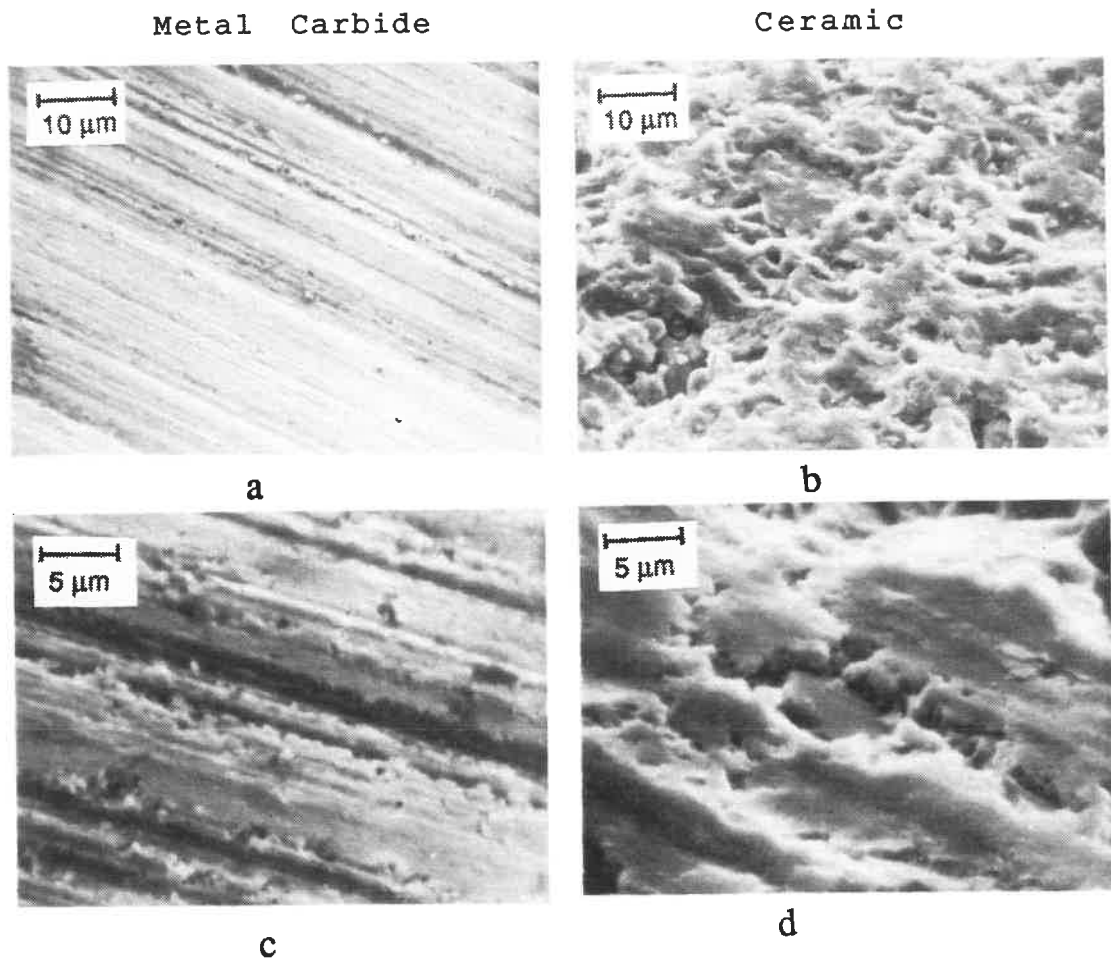


Fig.4

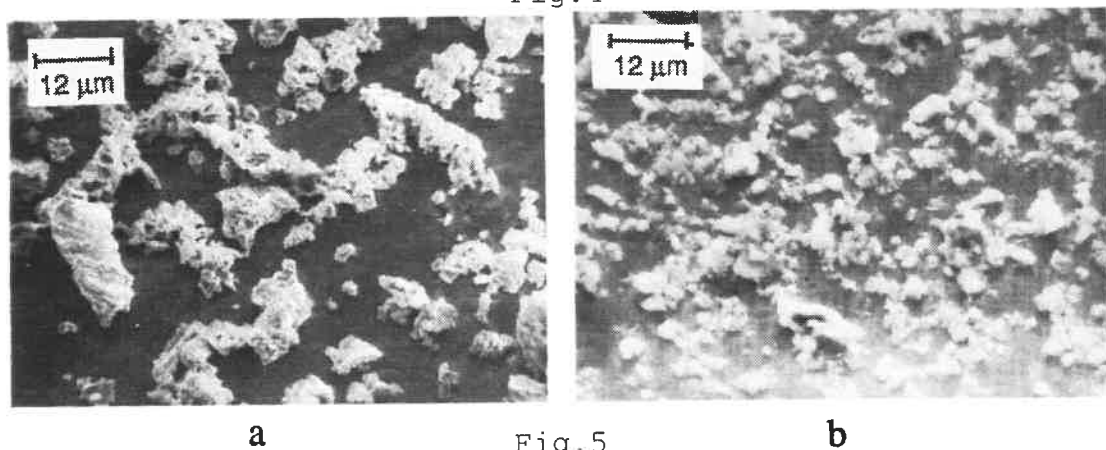


Fig.5

ADF AND NEURAL NETWORK MONITORING OF ABNORMAL GRINDING CAUSED BY LOADING

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Key words; grinding condition, aluminum alloy, adaptive digital filter, neural network, genetic algorithms

1 Introduction

When grinding aluminum alloy, chip loading deteriorates the grinding wheel surface, and chatter marks appear on the surface of a workpiece as wheel wear proceeds. In this paper, we propose two methods of monitoring abnormal grinding conditions: application of an adaptive digital filter (ADF) and a neural network developed by means of genetic algorithms (GA).

Chips adhere at random locations on the wheel surface, and the accumulated chips cause wheel imbalance. The revolution of an unbalanced wheel leads to vibration of the spinning shaft. In this investigation, a measuring apparatus is originally contrived in order to directly sense abnormal grinding conditions from the shaft vibration (Fig.1)¹⁾. The vibration signal from an acceleration sensor magnetically attached to a bearing fitted to the wheel shaft is statistically analyzed.

Changes manifested in both an index calculated from ADF and the neural network output are discussed; these changes are compared with changes in the wheel-wear process, as evidenced in grinding forces and the chip-loading ratio.

2 Experiments

A C46LmV wheel ground the surface of a workpiece made of material (Al-19%Si alloy) that is difficult to work with. Grinding conditions are shown in Table 1. Grinding forces measured by an octagon ring are shown in Fig.2. On the macroscopic level, the entire process with respect to grinding capability is classified into three stages²⁾ whose two transition points are at the 50th and the 250th cycles of grinding. In stage I, the tangential grinding force rises rapidly. In stage II, it increases gradually. In stage III, it decreases slowly.

Symptoms of chatter are visually observed at the 40~45th cycle of grinding, and chatter marks appear at the 50th (Photo.1). Some groups of stripes or squares coalesce to generate chatter

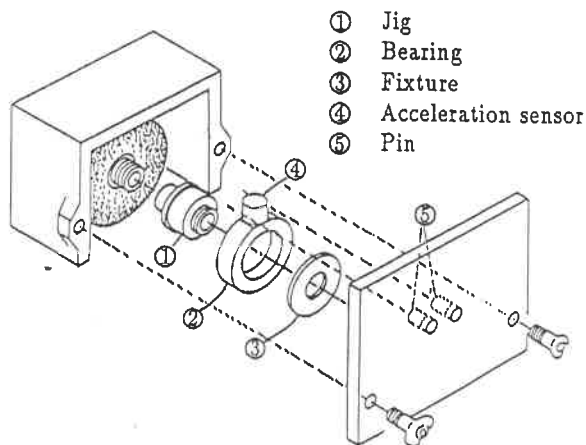


Table 1 Grinding condition

Classification	Condition
Size of work pieces	8×8×40mm
Wheel speed	1900m/min
Work speed	6m/min
Depth of cut	5μm
Dressing	10μm×8, 0.6m/min
Grinding fluid	Dry
Grinding method	Plunge Grinding
Wheel	C46LmV
Size of Wheel	φ1800×φ31.75×13mm
Grinding Machine	Okamoto PFG-450C

Fig.1 Attachment of the acceleration sensor

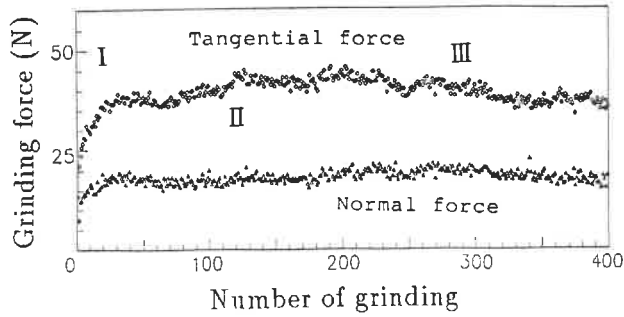


Fig.2 Grinding forces

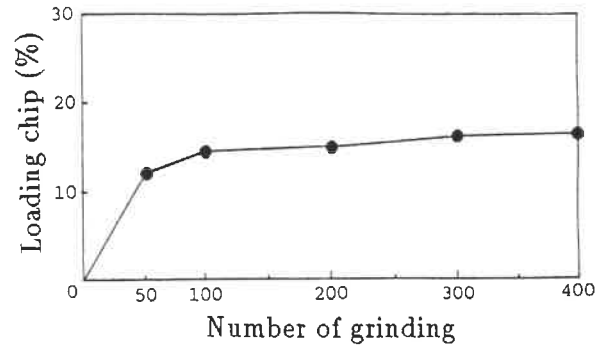


Fig.3 Loading chip area

at the 50th cycle of grinding. Photographs of the same point on the wheel surface show the chip growth. The total visual area of chips on the wheel remains constant after the appearance of chatter (Photo.2 and Fig.3). The constant chip loading suggests that chips larger than a threshold size fall off and new chips adhere to the surface of the wheel. Although the observed chip loading area does not increase, vibration of the wheel shaft increases as grinding proceeds.

Measurement of the grinding shaft vibration shows the existence of a sub-transition point at the 120th cycle in stage II. However, the measurement of grinding forces does not show this (Figs.2 and 6).

3 ADF and Signal Pattern Index

ADF³⁾ identifies an input signal adaptively by smoothing its slight changes. The parameters of ADF characterize the signal performance. The scheme of ADF used here is illustrated in Fig.4, where x_j and y_j denote the input and the output of ADF, respectively, ε its difference and z^{-1} a delay operator which satisfies $z^{-1}x_n = x_{n-1}$. The subscript j indicates a time period. An equation relating the input and the output of ADF is given, where the degree of the filter is $(N + M + 1)$, by

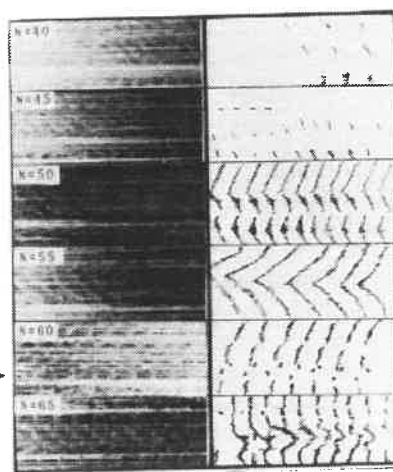
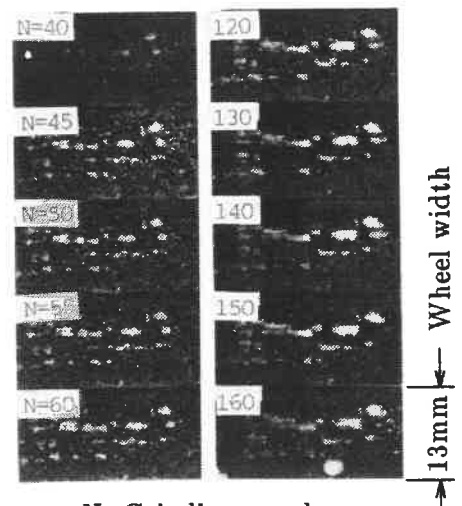


Photo. Sketch

N: Grinding number

Photo.1 Chatter mark



N: Grinding number

Photo.2 Chips

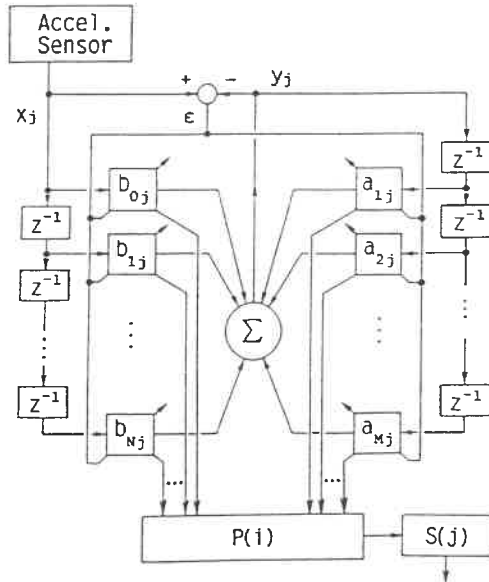


Fig.4 Structure of ADF

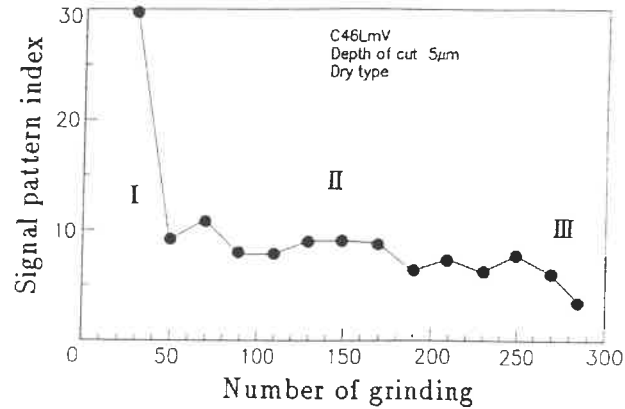


Fig.5 Signal pattern index

$$y_j = \sum_{k=0}^N b_{kj} x_{j-k} + \sum_{k=1}^M a_{kj} y_{j-k}. \quad (1)$$

The steepest-descent method adjusts ADF parameters adaptively by minimizing the error ϵ . $P(i)$ and $S(j)$ are calculated by the following equations (3) and (4), respectively.

Letting a parameter set in ADF at an instant j be

$$\Phi_j = [b_{0j}, b_{1j}, \dots, b_{Nj}; a_{1j}, a_{2j}, \dots, a_{Mj}], \quad (2)$$

$$P(i) = \sum_{n=0}^N (b_{n0} - b_{ni})^2 + \sum_{m=1}^M (a_{m0} - a_{mi})^2 \quad (3)$$

is derived as a scale expressing the difference in corresponding parameters of the model for the initial condition $j = 0$ and at an instant i . When ADF adapts its parameters to a time series at $j = k$, by using q -pieces appearing thereafter, the signal pattern index $S(j)$ is set as follows:

$$S(j) = \log \left[q / \left\{ \sum_{i=k+1}^{k+q} P(i) \right\} \right]. \quad (4)$$

In this investigation, $N = M = 15$. The ADF signal pattern index indicates that the stage change caused by abnormal vibration occurred due to the combined wear on the workpiece and the grinding wheel (Fig.5).

4 Neural Network

The neural network is a mathematical and connectionist model of the nervous system⁴⁾, where many neurons form a network with informational links. A neural network with well-controlled connections has certain intellectual abilities, for example, pattern recognition. Several forms of neural network have been proposed. We adopt a three-layered neural network: 21 input units-10 hidden units-3 outputs, where output characteristic functions of neurons are sigmoid.

The input data are given in the form of stochastic characteristics; the vibration signal in one revolution of the wheel is A/D-converted after passing through a band-pass filter, and transformed into the normalized peak-distribution map.

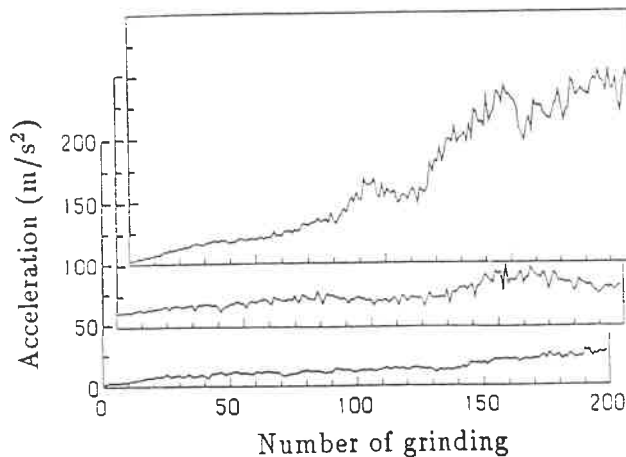


Fig.6 Vibration of grinding shaft

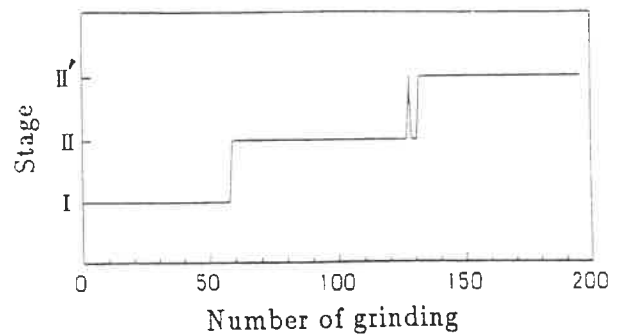


Fig.7 Judgement by neural network

The linkage weights of neurons must be trained to fit the purpose which is to warn an operator of the beginning of chatter using vibration signals and grinding numbers. In this investigation we adopt genetic algorithms⁵⁾ to optimize the neural network. The backpropagation supported by the steepest-descent method was avoided, because it involves a risk of the capture of local minimum points. The number of sets of genetic codes is 20. The evolution is shown by computer graphics, and is stopped when conversion is recognized. The weight set of the genetic code with the least error is adopted for the neural network for on-line stage recognition.

The neural network responds sensitively, and occasionally yields undesirable output. Adoption of the time-series-majority decision process allows control of the unsteadiness of the neural network. A software ratchet is set to maintain the irreversibility of the stage transition.

Changes of vibration as indicated by acceleration (rms) are shown in Fig.6. Two grinding processes train a neural network, and one process verifies this method. Figure 7 shows the final output of the neural network, which exhibits a general response to grinding processes.

5 Conclusion

Sensing vibration is simpler and easier than measuring grinding forces, surface roughness of the workpiece and chip-loading ratio on the wheel. The same vibration signal can be used in both the ADF index and the neural network. The ADF index is effective for macroscopic monitoring. Neural network monitoring is effective for automatically sensing such drastic change in grinding conditions as chatter, and for sensing the sub-transition points by training if necessary.

These methods should be verified using reliability tests.

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MICROSTRUCTURAL ANALYSES OF CHIP FORMATION IN POLYCRYSTALLINE CERAMICS GRINDING

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ABSTRACT *In this paper, the microstructure of a piece of polycrystalline alumina was analysed using a polarizing microscope for grain shape and size, pore distribution, boundary and impurity. The damaged surfaces of the ceramic scratched by fly-milling diamond abrasives were investigated using a SEM. The mechanisms of chip formation under the influences of internal stresses on the boundaries, impurities and pores in the crystals and along the boundaries were studied.*

INTRODUCTION

Many kinds of fine ceramics have been applied in many fields as new structural materials. It is well known from a large number of papers that diamond wheel is a very effective and indispensable wheel type for high efficiency grinding of fine ceramics[1].

Reports claim that in diamond grinding of brittle ceramics, much of the material removal is by brittle fracture[2-5]. Crack growth in brittle materials is accompanied by non-linear and non-elastic deformation which occurs in a process zone. Related to this phenomenon are crack closure effects and an irreversible elongation of the specimen. These effects are due to the volume increase of the damage or process zone as a consequence of microcracking in front of the macrocrack. Generally microcracking is initiated by internal stresses which are related to limited volume and shape changes during fabrication or stressing. Internal stresses originate from residual strains which are due to a thermomechanical mismatch of adjacent grains in facets or to phase transformation. The fracture characteristics are under the influence of the microstructure, and in particular on the absolute value of grain size as well as the grain and pore size distribution. The microstructural features, together with the elastic properties and the residual strain, determine the mechanical properties and the size of the damage zone[6].

The chip-producing mechanism is related to

the material properties and the operation conditions. The mechanical behaviors of ceramics greatly depend on the chemical structures and microstructural features of ceramics.

The aim of this paper is to provide a description of the microstructural analyses of chip formation in polycrystalline ceramics grinding.

MICROSTRUCTURAL CHARACTERISTICS OF CERAMIC WORKPIECE

The microstructure of ceramics is one of the most inherent factors which determine the mechanical behavior of a material and the size of the damage zone. The microstructure of the ceramic workpiece was analysed using a polarizing microscope. The analytical results of the microstructural features are shown below.

Grains The ceramic workpiece was formed from a polycrystalline alpha-phase alumina with 95% purity. Most of the grains are column-shaped. The average grain size is about $42\mu\text{m}\times 15\mu\text{m}$ and the maximum size is about $96\mu\text{m}\times 24\mu\text{m}$ as shown in Figure 1.

Boundaries In the workpiece, there was a vitreous-bonded oxide which contained about 5% of amorphous phase at the grain boundaries. When crack nucleation occurred, the cracks propagated along grain boundaries as long as the stress required for propagation was maintained. Because there are plenty of defects

at the grain boundaries, diffusion occurred more easily at the grain boundaries than that inside the grains. The grain boundaries are the important channels of material movement. The grain boundaries have a great influence on the physical properties of the ceramics, especially on the mechanical strength.

Pores Ceramics, because they are typically sintered from powder compacts, and after fabrication processes, unavoidably some pores remain. In Figure 1, there are some pores with a size of a few microns inside the grains and at the grain boundaries. The distribution, amount and size of the pores play an important role in fracture.

FLY-MILLING EXPERIMENTS BY SINGLE-POINT DIAMONDS

The experiments were performed on a M7120A Surface Grinding Machine at a speed of 2800 rpm. Instead of a grinding wheel an aluminum disk with a diameter of 180mm was used with a diamond abrasive grain mounted on the periphery, as shown in Figure 2. The depths of the cut were 0–10 μ m.

The polished ceramic workpiece was ground with the single-point diamonds. The damaged surfaces were examined in a SEM. The surface was coated with a thin layer of gold-palladium to overcome the charging problems normally encountered in the SEM with non-conducting materials. Stereo pictures were generally taken to obtain information.

During the grinding process, around the edges of the grooves, the main damage to the polycrystalline ceramic was due to transgranular fracture, as shown in Figure 3, and intergranular fracture, as shown in Figure 4. It was illustrated that, when the polycrystalline ceramics were ground, the removal mechanism was brittle fracture.

MICROSTRUCTURAL ANALYSES OF FRACTURE IN CERAMICS GRINDING

When machined, ceramics can deform through brittle mechanisms, i.e., material

removal is accomplished through the propagation and intersection of cracks. The microcrack porosity correlates with the microstructure. The brittle mechanisms can be given explanation through an analysis of actions between the stress (elastic and/or plastic) fields around the crack tips and the defect structures of ceramics.

Influence of Residual Stress on Grain Facets during Fabrication

Microcracks in single phase brittle polycrystals are a direct consequence of localized residual stresses[7]. These stresses derive from anisotropy in the thermal expansion properties of the material. Such anisotropy is typical of non-cubic materials. Thermal expansion anisotropy results in residual stresses with a wavelength of the grain facet length, which oscillate between tension and compression along adjoining facets. Figure 5 shows the location of microcracks with respect to the residual stress on the grain facets. The degree of thermomechanical mismatch of grains correlates with the grain size of ceramics. In large grained ceramics, the mismatch degree is high. Basically, crack nucleation is initiated by internal stresses. And then crack growth is developed by the action of external forces. In the grinding process, ceramics were impulsed by the diamond abrasives and cracks were nucleated and brittle fracture occurred mostly along the grain boundaries.

Influence of Impurities

According to the analytical results of the ceramic microstructure, we knew that, at the boundaries, there were extrinsic second phase impurities. Because the strain rates of the impurities were different from that of the basic crystals, when the external forces acted, ceramic polycrystals slid easily at the weak junctions. The crystal slides at the boundaries produced the gaps, as shown in Figure 6. And the stress concentration was induced at the gaps. Microcracking was developed by stress concentration and transgranular fracture occurred.

Influence of Pores

The pores are created during the fabrication

and the sintering processes. Microscopical observation showed that the pores existed at the boundaries and inside the crystals. When the pore is located at the juncture of three crystals, the pore can become the failure origin due to the stress concentration at the pore. When an applied force was activated, transgranular fracture occurred, as shown in Figure 7. The crack was initiated from the pore, and grew and developed along the boundaries and perpendicular to the direction of stress. When the pores remained inside the crystals, the pores played an important role in intergranular fracture.

CONCLUSIONS

In ceramics grinding process, the material removal is mainly by brittle mechanism, i.e., in brittle ceramics, grinding introduces transgranular and intergranular fracture into the machined surface.

Brittle mechanism of ceramic chip formation correlates with the microstructure of the ceramic workpiece. Residual stresses on the grain facets during fabrication was one of the causes that produced transgranular fracture. When the diamond abrasives were acting, ceramic polycrystals slid easily at the weak junctions where there were some extrinsic second phase impurities which strain rates were different from that of the basic crystals. Pores created during fabrication and sintering processes are one of the important failure origins in ceramics. Pores existing along the grain boundaries induced transgranular fracture. Pores remaining inside the crystals caused intergranular fracture.

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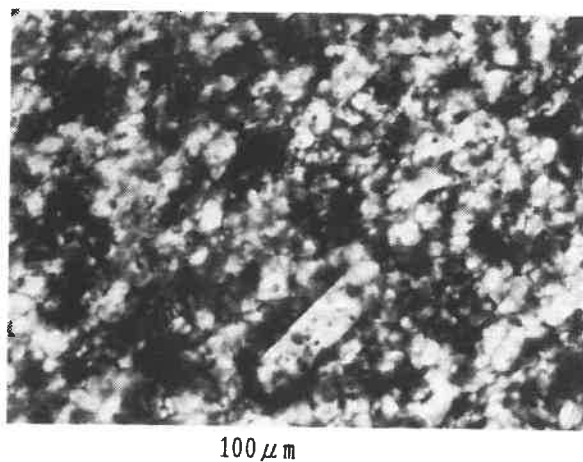


Figure 1. Microstructure of ceramic workpiece

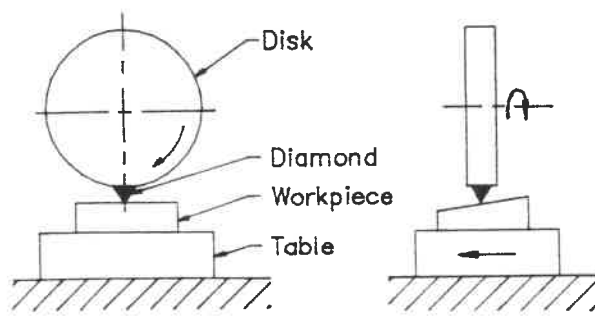


Figure 2. Single-point diamond grinding experiment

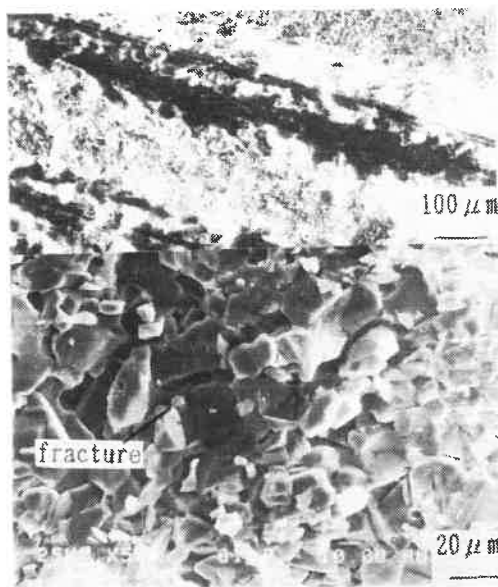


Figure 3. SEM micrograph showing transgranular fracture

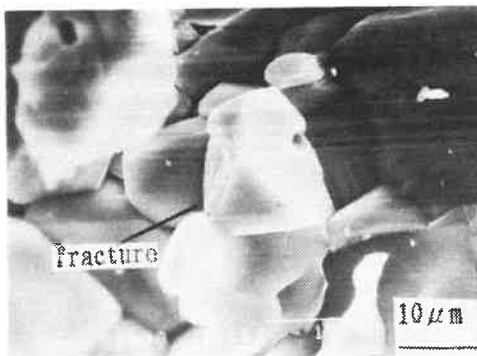


Figure 4. SEM micrograph showing intergranular fracture

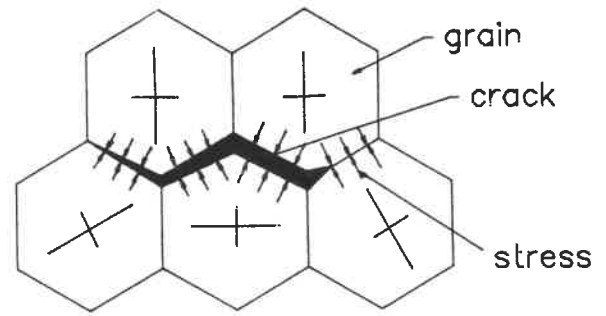


Figure 5. The location of microcracks with respect to the residual stress on grain facets

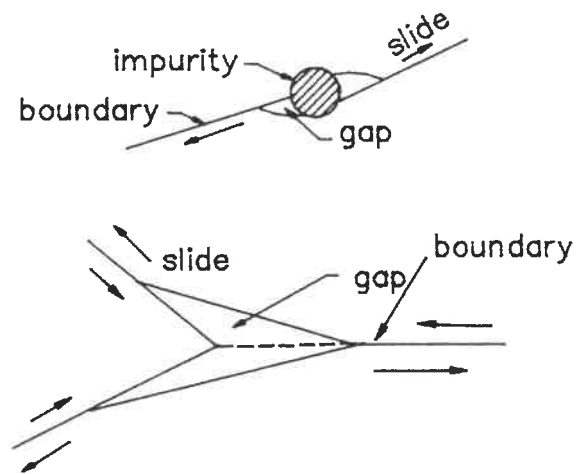


Figure 6. The gaps at boundaries

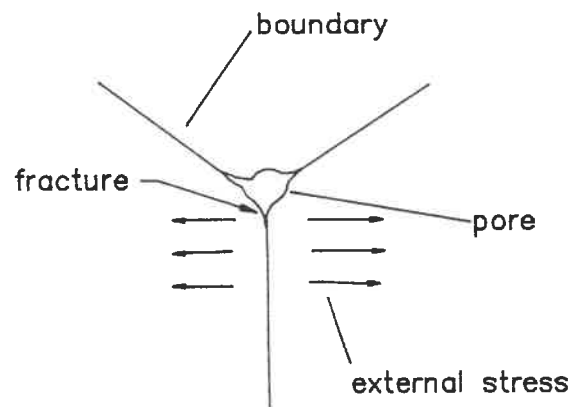


Figure 7. The pores at boundaries

Fine Ceramics Grinding Temperature

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1. Introduction

Diamond grinding wheels are usually employed to finish machine fine ceramics, but there are some problems such as surface damage and wheel wear. It is inevitable that the grinding temperature must be determined in order to analyze material removal mechanism, surface damage, and wheel life. Thermocouple method can detect grinding temperature which provides useful informations to analysis of grinding process, thereby is often utilized for measuring metal grinding temperature. However, thermocouple method has never been applied to detect fine ceramics grinding temperature, because this method is based on electric conductivity of workmaterials. Although $\text{Al}_2\text{O}_3\text{-TiC}$, widely used for tool materials, is absolutely a kind of fine ceramics, it also has electric conductivity. Therefore $\text{Al}_2\text{O}_3\text{-TiC}$ has been employed to assess the fine ceramics grinding temperature as the workmaterial and effects of two kinds of grinding temperature on the material removal mechanism has been analyzed.

2. Experimental procedure

Grinding temperature can be measured by thermocouples which can detect thermoelectromotive force between $\text{Al}_2\text{O}_3\text{-TiC}$ workmaterials and a chromel wire in grinding. Therefore, a 0.1 mm diameter chromel wire inserted between two mica platelets is clamped by two $\text{Al}_2\text{O}_3\text{-TiC}$ blocks being workpieces. Fig.1 shows experimental setup. A surface grinding machine is employed for machining experiments. Fig.2 shows typical grinding temperature curves measured. Each pulse in Fig.2 appears to represent the grinding temperature by individual abrasive particles. The highest point is defined as the fleeting temperature ' θ_g '.¹⁾ Similarly, temperature on the surface with contact to the grinding wheel, when individual abrasive particles are not machining the workpiece, is represents contour curve of wave bottom, namely envelope. A summit of the envelope which means highest temperature of the surface temperature is defined as longer acting temperature ' θ_m '.¹⁾ The intervals of the pulses indicates the length between each successive cutting point spacing, thus the average interval of successive cutting grains of the wheel used for this experiment was measured to be 6.7 mm.

Grinding forces are simultaneously measured by quartz transducers with high sensitivity and rigidity, thereby output of the transducers is recorded by a high frequency recorder. Table 1 demonstrates the physical properties of the workmaterials. Table 2 shows the grinding conditions.

Bearing steels are also employed for the experiment as a typical metal work material.

3. Effects of interference depth on grinding temperature

3.1 Effects of setting wheel depth of cut Δ

Fig.3 shows the variation in grinding temperature with setting wheel depth of cut. In Fig.3 θ_g in bearing steel grinding rose from 70°C to 530°C with an increase of Δ , while θ_g in $\text{Al}_2\text{O}_3\text{-TiC}$

grinding similarly rose with increasing Δ , exceeding 600°C at $\Delta=5\mu\text{m}$, which temperature was much higher than the former. Then θ_g attained a constant value around $\Delta=15\mu\text{m}$. This temperature rising mode could not be found in bearing steel grinding.

In Fig.3 θ_m in bearing steel grinding, being approximately 50°C lower than θ_g at every Δ value, rose with increasing Δ . The experimental results indicate that θ_m varied with θ_g accordingly. However, θ_m in $\text{Al}_2\text{O}_3\text{-TiC}$ grinding linearly rose with an increase of Δ , thereby a close correlation between θ_m and θ_g could not be found. Although θ_g in $\text{Al}_2\text{O}_3\text{-TiC}$ grinding was much higher than in bearing steel grinding, no significant difference of θ_m between $\text{Al}_2\text{O}_3\text{-TiC}$ and bearing steel grinding could be recognized. Rate of heat flow into the workpieces in $\text{Al}_2\text{O}_3\text{-TiC}$ grinding, therefore, seems to be low.

3.2 Effects of velocity ratio v/V

Fig.4.1 shows the variation in grinding temperature with velocity ratio v/V under three kinds of grinding speeds and constant setting wheel depth of cut conditions. For bearing steel, θ_g rose, on the other hand, θ_m lowered with an increase of v . Both values became higher with increasing V . In the case of $\text{Al}_2\text{O}_3\text{-TiC}$, θ_m showed similar trends to bearing steel, but a valley could be found in each θ_g curve around $v/V=1\times 10^{-2}$ at any table speed.

3.3 Effects of grinding temperature on material removal mechanism and surface damages

The variation of grinding conditions such as Δ , v/V are equivalent to changing the maximum chip thickness t_{max} and chip length l_c . That is to say, larger Δ values represent an increase of t_{max} and l_c leading to enlarging of chip size and v/V can control only t_{max} value.

It is well known that almost all energy dissipated in chip formation is converted into heat. In ductile mode grinding, the energy should be dissipated according to volume of material removal. Hence, θ_g appears to rise proportionally to the volume of the material removal, then reached nearly constant values at the inflection point.

In brittle mode grinding the energy per unit removal volume seems to be smaller than in ductile mode grinding, because the energy required in chip formation mainly composed of crack generation energy.²⁾ The transition point from ductile to brittle mode in brittle material grinding is earnestly discussed among brittle material machining engineers and is supposed to be dependent on interference depth.

If the material removal mechanism did not change, the grinding force ratio should hold a constant grinding force ratio. In fact, the ratio drops at the inflection point, where t_{max} shows a $0.6\mu\text{m}$ value. Fig.4.2 shows variation in θ_g with t_{max} . The figure indicates that each θ_g curve has a valley at $t_{\text{max}}=0.5\text{-}0.6\mu\text{m}$, which seems to be inflection point, as well as Fig.4.1.

The actual wheel depth of cut Δa becomes smaller with an increase of v , which led enlarged t_{max} and reduced l_c . It can be found that θ_g is controlled by t_{max} and l_c , one of which can govern the grinding mode. At the inflection point factors such as t_{max} and l_c may dramatically interchange as dominant factor. Although θ_g might be independent of θ_m , the former should have a close correlation to the latter as well as bearing steel grinding. The correlation may be based on friction heat in addition to low heat conductivity of ceramics. The experimental results suggest the transition from ductile to brittle grinding mode takes place at the inflection point.

Fig.5 shows SEM micrographs of the surfaces ground. The micrograph indicates grinding streaks which generated from plastic flow all over the surface of $\Delta=10\mu\text{m}$ grinding. While, a variety of chippings due to brittle fracture could be seen on $\Delta=30\mu\text{m}$ grinding surface. Lateral cracks around the chippings could also be observed. Transition of the ground surface texture exactly corresponds to the measurement results of fleeting temperature θ_g .

4. Effects of grinding fluid

The grinding fluids are widely used in grinding operation in order to lower grinding temperature and lubricate the interface, etc. The heat generated in grinding process is expected to be lowered by the fluids to a great extent. Fig.6 shows effects of the cooling action on the grinding temperature. The figure indicates the fluid positively lowered θ_g about 100°C . The fluid had significant cooling effects on θ_m that can be regarded as temperature of the surface ground. Actually, θ_m in wet grinding offered 100°C at larger $t_{\max}$, where θ_m value in dry grinding reached more than 350°C . In more than $\Delta=30\mu\text{m}$ grinding the workpieces could not be machined because of fracture of workpieces, while wet grinding positively produced chip formation process. This fracture generated in chip formation process may be caused by thermal stress associated with rapid heat cycle beneath the ground surface. If the fracture could not be seen, some micro cracks might have generated in the subsurface. It is inevitable to utilize grinding fluids in order to finish machine $\text{Al}_2\text{O}_3\text{-TiC}$ material with high quality.

5. Conclusion

Thermocouple method can be applied to detect $\text{Al}_2\text{O}_3\text{-TiC}$ ceramics fleeting and longer acting temperature in grinding. The fleeting temperature was much higher than in bearing steel grinding. However, no significant difference in the longer action temperature between them could be found. Experimental results show that rate of heat flow into the workpiece to the grinding heat was small in $\text{Al}_2\text{O}_3\text{-TiC}$ grinding. The fleeting temperature became higher with an increase of the material removal volume, then reached at a constant value over the inflection point, namely $t_{\max}=0.5 - 0.6\mu\text{m}$. This inflection point could be in agreement with the transition point from ductile to brittle grinding mode.

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Table 1 Physical properties of workmaterials

	$\text{Al}_2\text{O}_3\text{-TiC}$	Bearing steel
Specific gravity	4.3	7.8
Bending strength MPa	900	—
Vickers hardness	2000	743
Thermal conductivity cal/cm · sec · $^\circ\text{C}$	0.05	0.11

Table 2 Grinding conditions

Grinding method	Surface grinding, Down cut	
Wheel type	100/120 Synthetic diamond, Metal bond, Concentration 125	
Wheel speeds : V_s m/min	1600	1000 - 1600
Table speeds : V_w m/min	10	5 - 20
Wheel depth of cut : Δ μm	5 - 30	10
Grinding fluids	None or Soluble type	

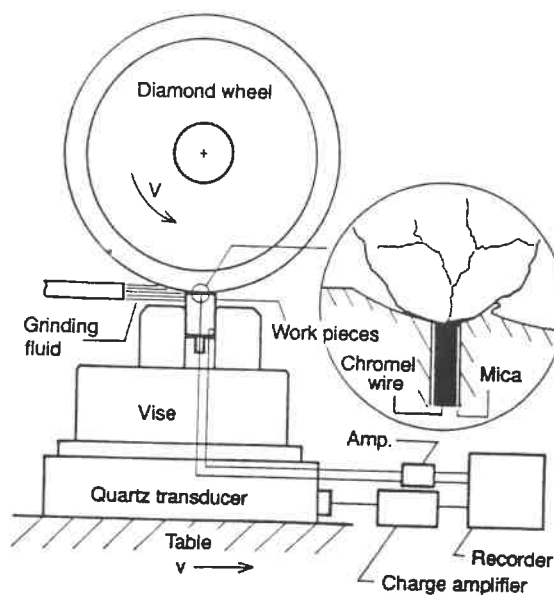


Fig.1 Temperature measurement device using thermocouples

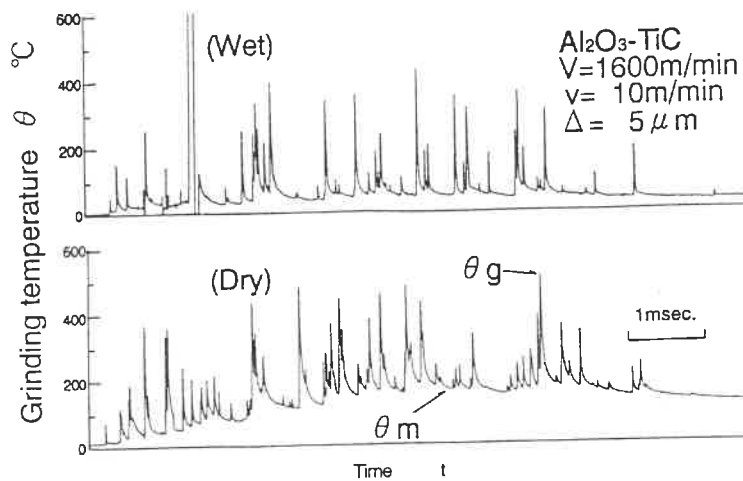


Fig.2 Typical temperature curves measured

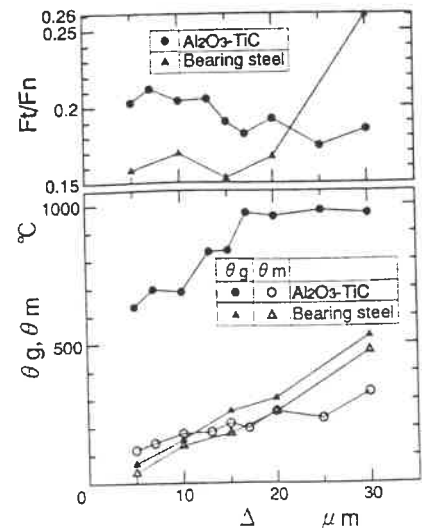


Fig.3 Effects of wheel depth of cut

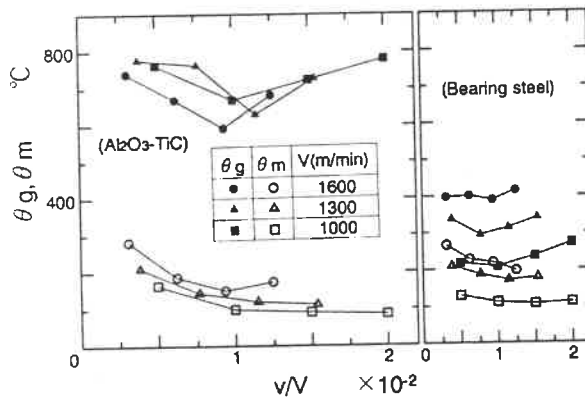


Fig.4.1 Effects of velocity ratio on grinding temperature

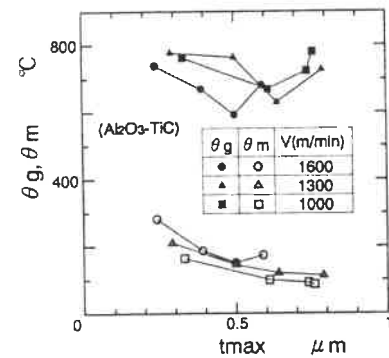


Fig.4.2 Effects of t_{max} on grinding temperature

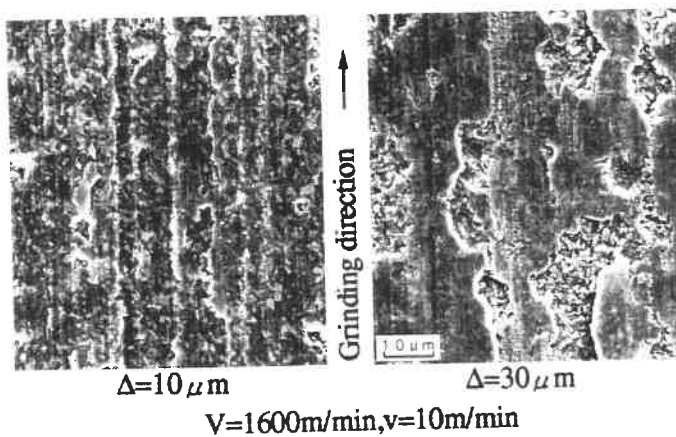


Fig.5 SEM micrographs of ground surfaces

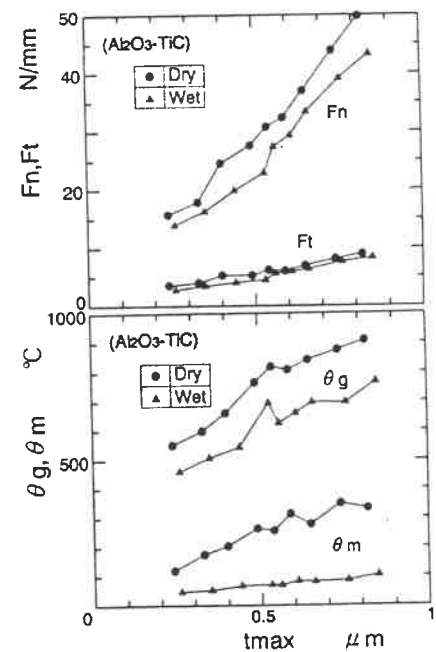


Fig.6 Effects of t_{max} on grinding temperature and force

PRECISION GRINDING REGIME OF ADVANCED CERAMICS

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1. Introduction

In precision grinding research, there is a new term used to describe the material-removal mechanisms, namely, 'ductile regime' grinding of brittle materials [1-3]. A hypothesis is advanced by some researchers, according to which all materials regardless of their hardness and brittleness, will undergo a transition from a brittle to a ductile machining region below a critical depth of cut. These researchers appreciated that by controlling a stiff, accurate grinding apparatus so that it has an exceptionally small scale of material-removal, brittle materials can be ground in a ductile manner. As a result, brittle workpiece can be machined in a deterministic process while producing surface finishes characteristic similar to those achieved in non-deterministic, inherently ductile processes such as lapping and polishing. The argument for 'ductile regime' grinding has been based on energy consumption in different material-removal processes. The energy required for brittle fracture is proportional to the square of a characteristic dimension that can be the depth of cut, while for plastic deformation to the cube of the characteristic dimension, resulting in the energy ratio of plastic flow to fracture proportional to the characteristic dimension. As the scale of machining decreases, plastic flow becomes an energetically more favorable material-removal mechanism [1, 2], the material-removal mechanism at the decreased machining scale thus is in 'ductile regime'. In other words, if the depth of cut is below the critical value, the energy required to propagate cracks is larger than the energy required for plastic deformation, so that plastic deformation is the predominant mechanism of material-removal in grinding these materials. In contrast to the 'ductile regime', powder regime has been found in this research, based on which material-removal has been achieved. This research deals with material-removal mechanisms involved in precision grinding of advanced ceramics. As the first step of the

investigation, single-point grinding technique is applied to silicon nitride, silicon carbide, alumina and zirconia ceramics.

2. Stress Model in Contact Region

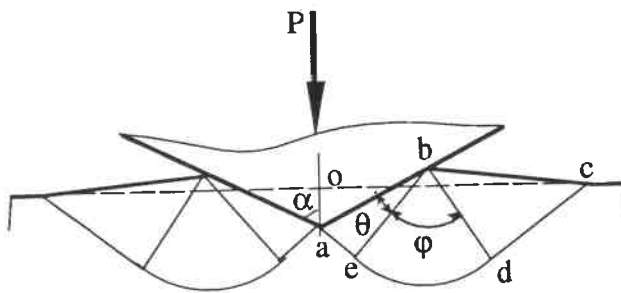
Slip-line field solution can be used to describe a stress model in single-point grinding of ceramics. The slip-line field solution is based on the assumption that the tool edge cuts the specimens, for example, along plane oa in Figure 1(a). It follows that this creates two new surfaces, which rotate about point a as the cutting edge penetrates into the work material. The material originally located at point o is thus relocated to point b and the material in $abcde$ is permanently deformed with a side-ways and upward motion. The irreversible deformation boundary is $cdea$. The state of stress within the regions of flow can be represented by a slip-line field, which is shown in Figure 1(b) by Mohr's circles with a constant radius k , where $k = Y/2$ as per Tresca criterion of yielding, and $k = Y/\sqrt{3}$ as per von Mises criterion. k and Y denote the values of the yield stresses in simple shear and in simple tension of the work material. The stress in the deformed regions can easily be found by Hencky's equations:

$$\begin{cases} \sigma - 2k\varphi = \text{const.} & \text{along an } \alpha \text{ line} \\ \sigma + 2k\varphi = \text{const.} & \text{along a } \beta \text{ line} \end{cases} \quad (1)$$

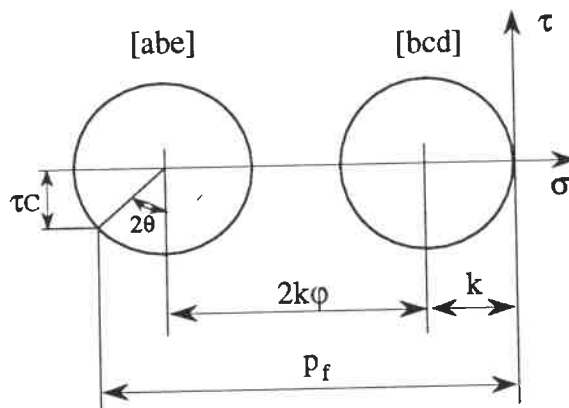
The pressure on the face of the cutting tool is now given by

$$P_f = k(1 + 2\varphi + \sin 2\theta) \quad (2)$$

In the region bcd , the hydrostatic component of stress has the value of k , shown by the centers of the circles. The hydrostatic component of stress increases as φ increases. It reaches the maximum value $k(1 + 2\varphi)$ in the region abe . The hydrostatic components in all three regions are superimposed by a shear stress k .



(a) Irreversible deformation caused by a cutting tool



(b) Moh's Circles

Figure 1 Slip-line field solution to the stress state of the single-point grinding of ceramics

4. Experimental

A series of single-point grinding experiments has been conducted. A precision grinding machine [4] was designed and constructed for the experimental work. This machine is equipped with an aerostatic spindle (speed range 300 - 15,000 rpm) and an aerostatic slideway. It is characterized by its high rotational accuracy (radial run-out 0.03 μm), high stiffness (35 N/ μm in structural loop), and high positioning resolution (0.5 μm).

In the single-point grinding tests, a ceramic workpiece was fixed on the periphery of an aluminum alloy disc mounted on the spindle. Workpiece was then shaped by a cylindrical grinding machine equipped with diamond grinding wheels, and polished to remove the damage layers caused by the grinding processes. The polishing was done by

diamond pastes with grain sizes ranging from 15 μm (for rough polishing) to 0.25 μm (for fine polishing). Single-point diamond, on the other hand, was fixed on the slideway through a piezoelectric force transducer. Diamonds with different inclusion angles (60°, 80°, 100° and 120° for different rake angle effects) and different nose radii (0.7 - 2.1 μm and 45 μm) were used.

The ground surface and subsurface layers were observed and measured by a scanning electron microscope and a surface profilometer. Since sub-surface damages are usually difficult to inspect, three destructive inspection (DI) techniques [5] were developed for the experiments. Different ceramics were chosen: hot-pressed silicon nitride and alumina, sintered silicon carbide, and slip-cast zirconia. The physical and mechanical properties of these work materials can be obtained in [6].

The single-point grinding tests were carried out in the following conditions:

Grinding speed:	1600 m/min
Depth of cut:	0 - 15 μm
Grinding length:	12 mm
Coolant application:	Dry and wet

Figure 2 shows SEM observations of surface and subsurface regions of alumina and silicon nitride after the single-point grinding tests. No cracks or chipping occurrence have been found in depth of cut less than a critical value (2.5 μm for silicon nitride, 2.0 μm for alumina, 0.5 for silicon carbide). However, when the depth of cut was larger than the critical values, cracks were observed in the subsurface layers. These cracks were usually covered by a pulverized layer or powder regime with a smooth surface shell as shown in the SEM photographs of Figure 2. The powder regime exhibited non-continuous and loose grain connections when the observation was prepared by the taper polishing technique, though it was difficult to be recognized by the fracture technique (Middle row in Figure 2). The loosely connected grains are easily pulled out by abrasive grains in a super polishing process. The grain was found to be finer in this powder regime by the fracture technique. Cracks, if induced, nucleated from the boundary of the powder regime.

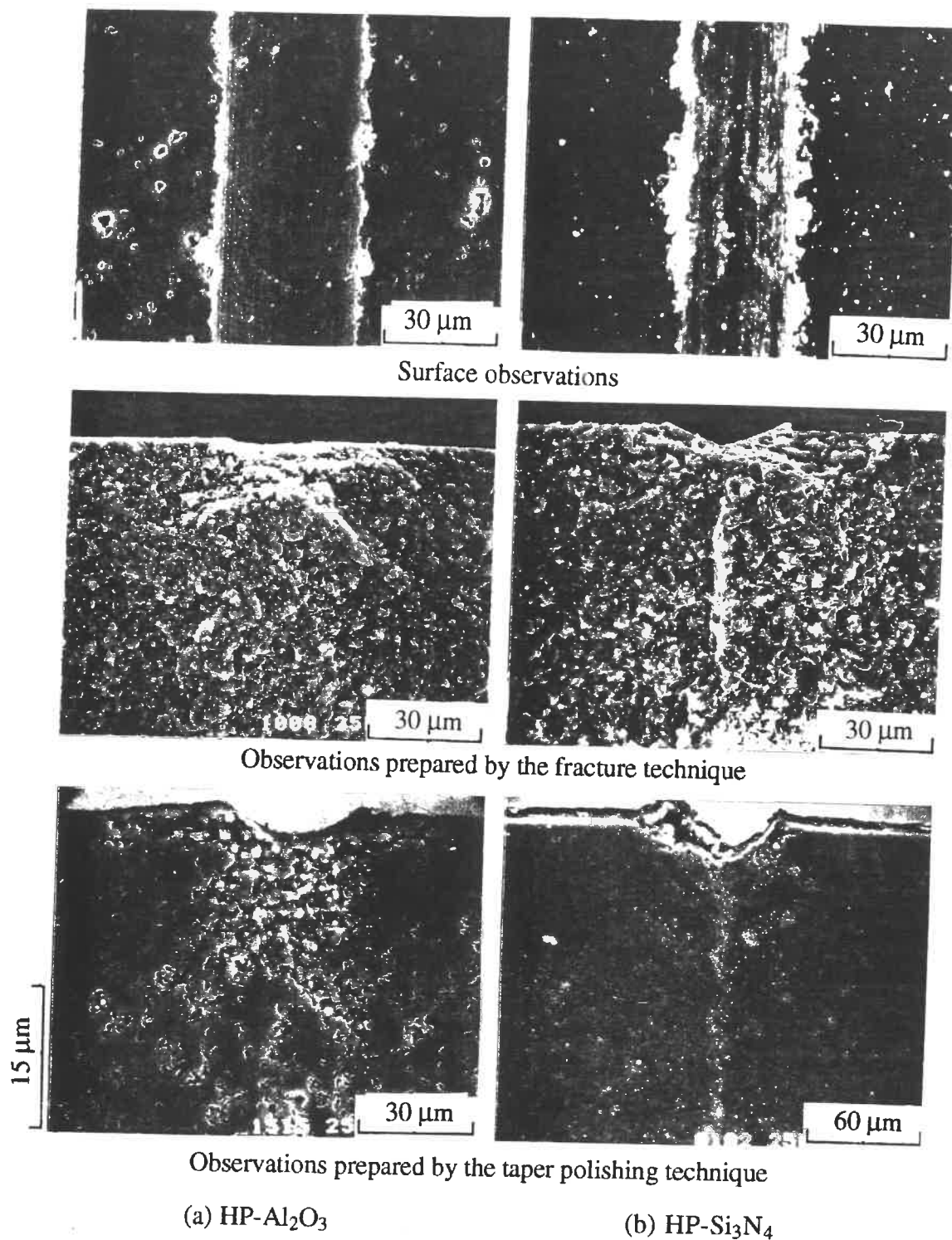


Figure 2 SEM observations of surface and subsurface regions of alumina and silicon nitride ground by single-point diamonds. The upper row shows the surface observations; the middle row the observations prepared by the fracture technique and the lower row the observations by the taper polishing technique. The 'powder regime' is observed in the lower row.

Figure 3 schematically shows the powder regime caused by a single-point grinding process. If the grinding load is small, or if the depth of grinding is within certain range, the material-removal is based on the powder regime, which results from pulverization of the work material, leaving a ground groove with a smooth surface. The powder regime results from the bond rupture of the material, for which the shear stress is responsible. The powder regime, if induced, is constrained by the bulk of the material within a half elliptical region. The constraint generates a hydrostatic pressure that may re-compact the pulverized material. Side flows of the pulverized material, on the other hand, are caused by the special stress state, forming material pile-ups on both sides of the ground groove as shown by the SEM photographs of Figure 2. Besides, due to the elevated temperatures in the contact region, the pulverized material could be re-sintered, which would also contribute to smoothing the surface of the ground groove.

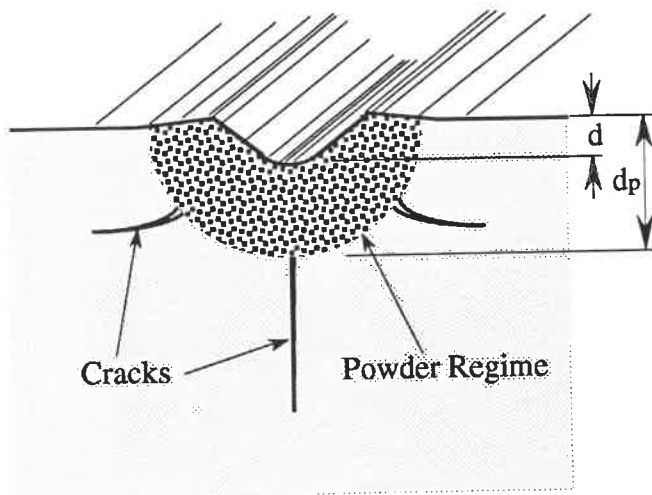
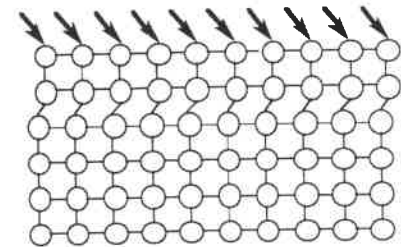


Figure 3 Schematic showing of powder regime induced in single-point grinding process

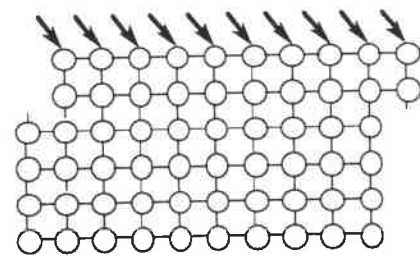
4. Discussion

In grinding process, the work material beneath and ahead of the cutting tool is deformed by a large hydrostatic compressive stress superimposed by a shear stress. This complex stress field causes the material to deform. The deformation can be in several stages as the stress level increases. From

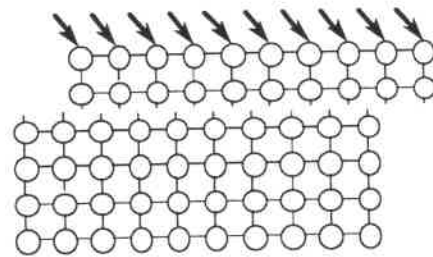
atomistic point of view, the progressive deformation of the material lattice is in the sequence of elastic deformation, dislocation and bond rupture, which can generate a powder regime for the material-removal. The processes of the lattice progressive deformations are schematically shown in Figure 4.



(a) Elastic deformation



(b) Dislocation



(c) Bond rupture

Figure 4 Lattice progressive deformations caused by shear stress

At the first stage in stressing the material, lattice distortion or elastic deformation occurs which is shown in Figure 4(a). As the shear stress increases, dislocation movement begins along certain slip planes. With the further increment of the shear stress level, the deformation is beyond the ability of the dislocation movement, bond rupture takes place, which results in the pulverization of the work material. The process of pulverization decreases the size of ceramic grains by breaking

If the brittle-ductile threshold given by [3] is employed, the critical effective force will be

$$(P_i^c)_{\text{critical}} = \gamma (K_c^4/H^3) f(E_w/H), \quad (1)$$

where K_c , H and E_w are respectively the fracture toughness, hardness and elastic modulus of the workpiece material, γ is a dimensionless constant, $f(E_w/H)$ is a weak function and $\gamma f(E_w/H) \approx 2 \times 10^5$ for most materials[3].

The effective forces reflect reasonably the following facts in grinding: (1) it takes into account the real distribution of the contact force over the grinding zone; and (2) it involves all interactions from active and inactive grains as well as loaded chips. The evaluation of N_e is based on experiments. A simple and realistic approximation is to equate N_e to the dynamic number of active grains proposed by Werner[6].

A problem of directly applying equation (1) should be addressed. As has been realised, the depth of grain cut is not a constant. Thus large effective forces take place only when the chip thickness becomes large. However, the part of the material where P^c exceeds the threshold will definitely be removed in grinding. It then follows that if the size of radial crack does not exceed the depth of wheel cut when this layer of material is removed, the fracture mechanism of chip formation can also give rise to a ground surfaces without micro-cracking. On the other hand, the effective forces on the effective grains at the tail part of the pressure profile are very small. These grains produce plastic flow rather than fracture and hence improve the quality of the surface finish.

Experimental study confirmed the above prediction. The bearing ratio(t_p) and surface roughness curves of Si_3N_4 with the same characteristics as that of a ductile material(bearing steel) were observed, see Fig.2, where grinding conditions were $t = 0.25$ mm and $v = 1.0$ mm/s. No surface micro-cracks were detected. The theoretical prediction of maximum effective force for this particular case is 54.8(N) which exceeds significantly the lateral cracking threshold of Si_3N_4 of 30.5 (N). In the calculation, measured values of $H=16$ (GPa) and $K_{IC}=5$ (MPa $\text{m}^{1/2}$) of the material were used.

The above results indicate the feasibility of achieving ductile-regime creep-feed grinding for brittle materials, which is of great importance to the precision engineering industry.

EFFECT OF ELASTIC MODULUS OF GRINDING WHEELS

The above discussion shows that the longer the pressure tail, the better surface could be achieved. Nevertheless, the distribution of pressure profile is determined by the deformation of the grinding wheel and the workpiece material. For a given workpiece material, a wheel with a smaller elastic modulus produces a longer contact length and hence a longer pressure tail. The mattress model(Fig.1) indicates that decreasing the elastic modulus of the wheel enlarges L_c , leading to more effective grains being included in the grinding zone and thus decreasing the maximum value of effective forces. As a result, it will lower the occurrence of cracks and reduce the depths of crack penetration. Consequently, for a given workpiece material, the selection of the elastic modulus of grinding wheels plays an important role in achieving ductile-regime grinding. However, the elastic modulus of a wheel is a strong decreasing function of grinding temperature rise[7]. It then follows that a large cooling rate

in the grinding zone can promote surface cracking. The experimental observations made by Eda[8] concerning the effect of elastic modulus of grinding wheels on the ground surface are therefore consistent with the present theoretical model.

To further verify the prediction and understand the importance of this effect, a special experiment was arranged. As is well known, the surface finish of a ground component is usually significantly influenced by the grain size of the grinding wheel: coarse grains produce poorer surface finish and vice versa. This is referred to as a grain size effect. In this experiment, a softer HA wheel with large grain size and a harder CBN wheel with fine grain size were chosen. The HA wheel had an average grain diameter of 250 μm , and that of the CBN wheel was of 88 μm . Both of them were vitrified bond but the CBN wheel had a steel hub and grinding with this wheel was performed under much higher cooling rates in the grinding zone. These arrangements guaranteed that the elastic modulus of the CBN wheel was much bigger. As expected, the results show that fine grains produced poor surface finish(Fig.3) and that the effect of elastic modulus could be much stronger than that of grain size.

CONCLUSIONS

- (1) The effective grains and effective forces are appropriate and practical parameters for studying the transition in grinding hard brittle materials. The significance of defining these parameters is that they take into account the real distribution of contact force over the grinding zone and involve all interactions on the wheel-workpiece interface.
- (2) The elastic modulus of a grinding wheel is an important factor which influences the surface finish significantly. The effect of elastic modulus can be explained clearly by the concept of effective force and the mattress model. From the point of view of micro-cracking restriction, high cooling rate in grinding zone is not always a good approach to produce fine surface.
- (3) A ductile-regime creep-feed grinding may be achieved by an optimal combination of grinding parameters. Further studies are required to clarify quantitative relations between grinding conditions and the control of grinding mechanisms.

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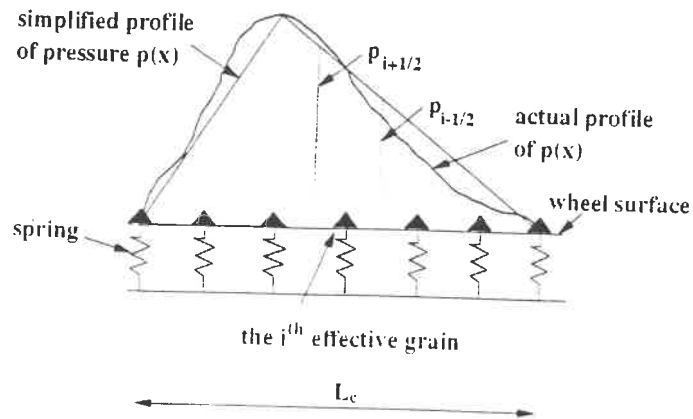


Fig.1 An example of the contact normal force over grinding zone

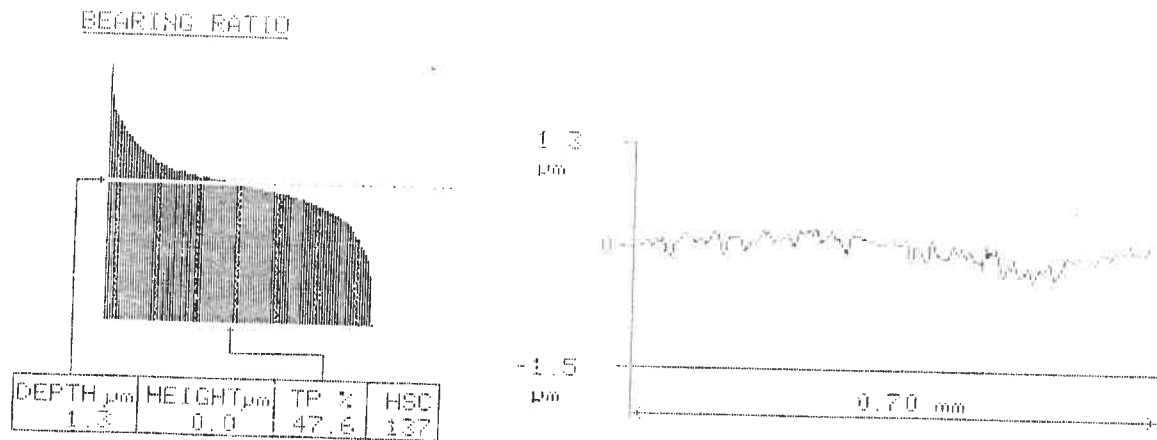


Fig.2 t_p and roughness curves of a ground Si_3N_4 surface

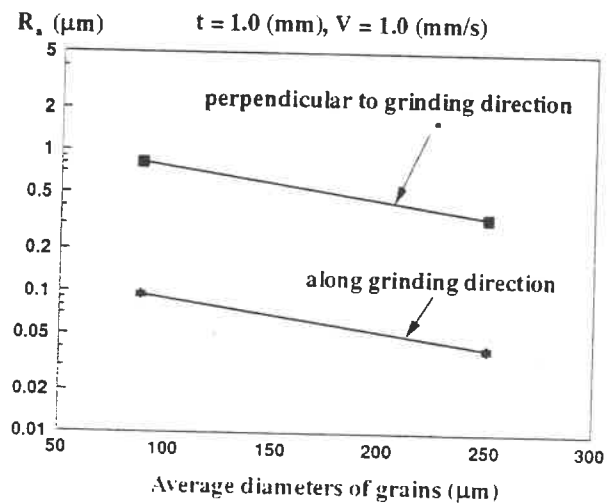


Fig.3 Surface roughness vs grain size

A New Truing/Dressing Method for Superabrasive Wheels

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1. INTRODUCTION

With a rapid growth in the use of superabrasive wheels, an important and difficult question we are facing with is truing/dressing operation. In ultra-precision grinding, truing accuracy and grinding performance are the key to improving the total quality of ground surface. Recently, several attempts have been made¹⁾. The diamond truing device, such as the rotary dresser, still is the most common tool to be used now. As truing a superabrasive wheel, the matter we must consider is the increased wear on the truing tool. The significant wear on the tool makes it difficult to generate the required geometry or profile, produces a wear flat on the abrasive particles and declines the grinding performance.

This paper proposes a new method for diamond/CBN wheels of any type bond material. The truing device employs a cup type Green Silicon Carbide (GC) wheel as the tool, offers both truing and dressing functions and produces better grinding performance post truing. The mechanism and results of the new device were investigated, and the application of the device to an arc profile truing was described.

2. The Cup-truer

A conventional abrasive, such as a WA (white corundum) stick, is usually used as a dresser to resharpen the superabrasive wheel after truing. The device shown in Fig.1 is designed to perform both truing and dressing functions. This equipment uses a cup type GC wheel as a truing tool which, unlike a normal

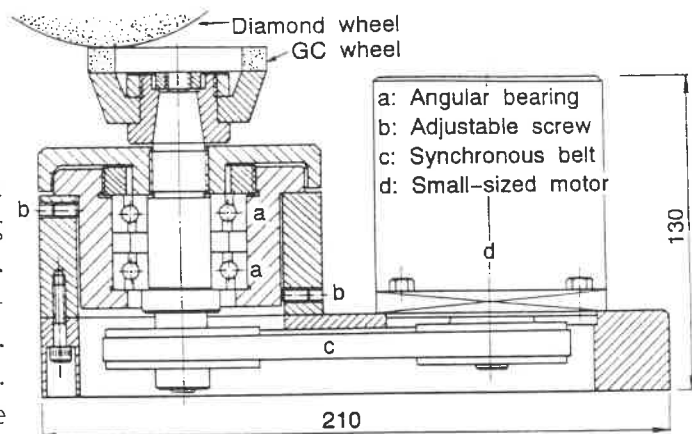


Fig.1 The GC Cup-truer

Table 1 The GC Cup-truer specifications

① Tool unit	
• Drive	AC speed-control motor
• GC wheel size	90mm ^D × 50mm ^H × 20mm ^T
• Speed	100~1200rpm
• Face camming	1μm/φ90mm(adjustable)
• Perpendicularity	0.03 degree(adjustable)
• Stiffness	50N/μm
② Size	
• Truer unit	210mm × 130mm × 100mm
• Controller	100mm × 100mm × 50mm

break truer, is driven by a small-size motor. The specifications of the device, referred to as the GC Cup-truer, is listed in table 1.

The construction allows the GC Cup-truer to infeed in a direction longitudinal to the wheel face, whereas other devices often require a traverse motion across the wheel face. Fig.2 shows the trued diamond wheel cross-section, compared to that of traverse motion truing. It is

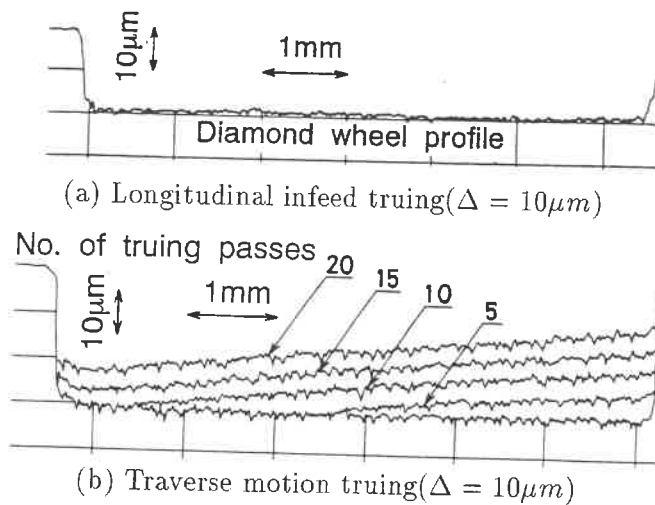


Fig.2 Cross-sectional profile

obvious that the infeed mechanism of the GC Cup-truer can avoid the influence of tool wear on the geometry accuracy, resulting in profile error less than $1\mu\text{m}$.

Far better is the wheel grinding performance post truing. Fig.3 shows the grinding force as comparing the GC Cup-truer with the other truing tools: (a) is that of a vitrified bonded diamond wheel at CaTiO_3 ceramic creep feed grinding, and (b) is that of a vitrified bonded CBN wheel at AlNiCo surface grinding. It is observed that diamond wheel trued by the GC Cup-truer requires the least grinding force, while the CBN wheel starts from a low grinding force and remains steady. The results clearly indicate that the wheel trued by the GC Cup-truer performs better than the wheels trued by most diamond tools which often damage the abrasive cutting edges and lead to a high grinding force. Furthermore, the other grinding performance of the CBN wheel, like surface roughness, is controllable by varying the grit size (or the hardness) of the GC wheel used.

Fig.4 is the SEM stereo pairs of the trued wheel surface, which demonstrate the truing mechanism made by the GC Cup-truer. (a) shows that of a metal bonded diamond wheel. The layer of the bond material is deep from the wheel peripheral surface, so that it is impossi-

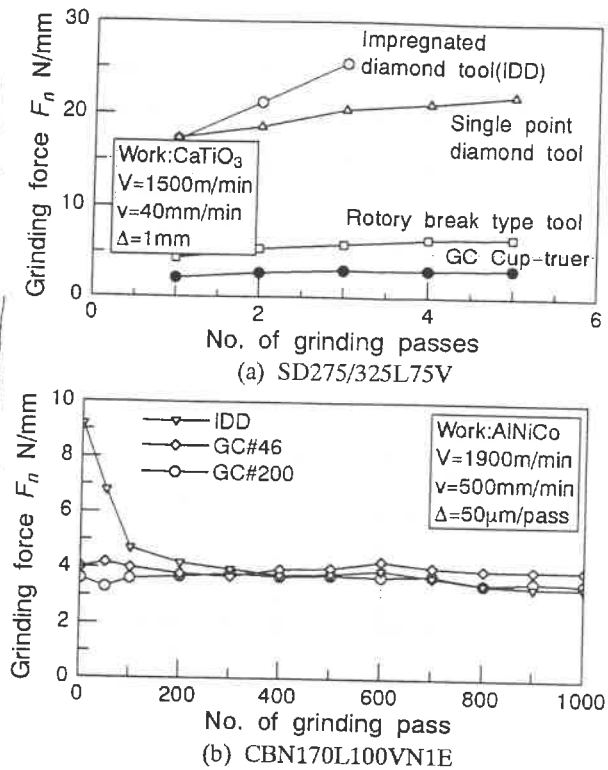
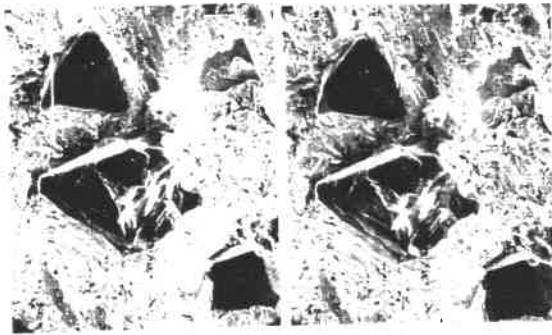


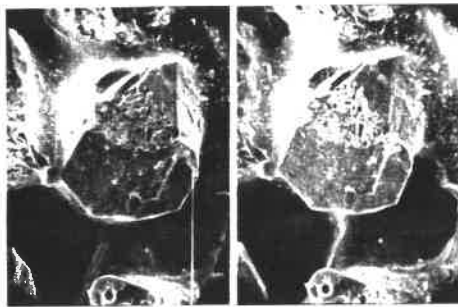
Fig.3 Comparison on grinding force

ble to contact directly with the truing tool. The scratch trace of GC grains, however, is observed on the bond face. It can be assumed that the GC grains are removed from the GC cup wheel during truing, enter into the diamond wheel and interfere the bond material. This "lapping" by the loose GC grains removes the bond material efficiently, but does not damage diamond particles. The effect obtained is equally applicable to the wheel of other bond materials or the CBN wheel. In case of using a coarse or a hard GC wheel to true CBN wheel, additionally, micro-cracks on the tip of grains are observed as shown as in (b), which creates multi cutting edges on one abrasive particle and leads to the stable grinding performance.

Fig.5 shows the influence of the supplied coolant flux on the truing ratio (the volume ratio of the trued wheel to consumed truer). According to the truing mechanism mentioned above, decreasing in truing ratio at excessive coolant supply can be explained quite naturally as due to that a large number of loose



(a) Diamond wheel



(b) CBN wheel

Fig.4 SEM stereo pair of wheel surface

GC grains were flushed out together with the coolant. Truing under a dry condition, contrary, the loose GC grains were scattered outward the truer face due to truer rotation, and then resulting in a low ratio. This result gives a good evidence for truing mechanism assumed above, and indicates that concentrating the loose GC grains on the truer surface reaches the maximum truing efficiency.

3. The Arc Truer

Making extended use of the straight cross-section formation with the GC Cup-truer, a convex profile can be formed by enveloping the straight lines. Fig.6 shows a new type truing/dressing device (i.e the Arc Truer) for a cylindrical grinding machine. The assembly is made up of four basic movements: ① a rotation unit for the truing tool, ② an infeed mecha-

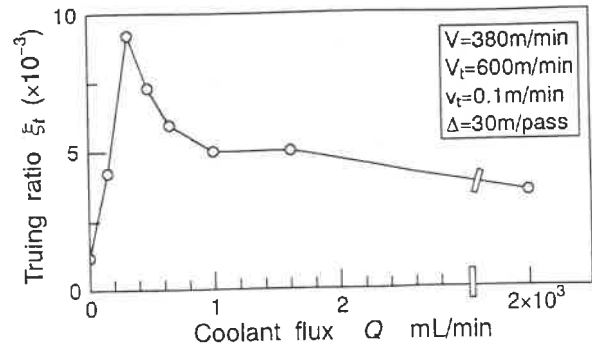


Fig.5 Influence of coolant supply

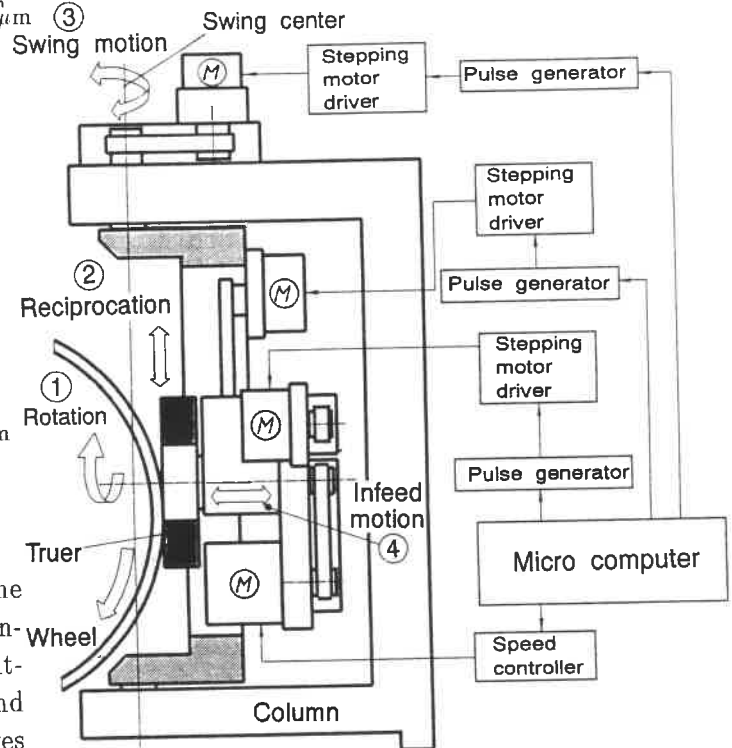
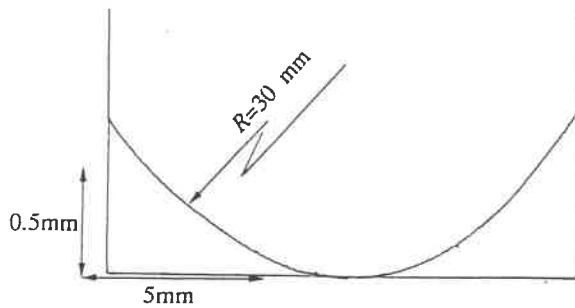


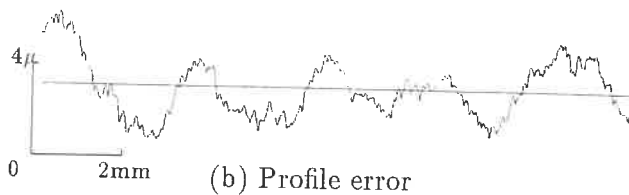
Fig.6 Schematic drawing of the Arc Truer

nism, ③ a crankshaft for vertical movement and ④ swing motion. All movements are controlled by a programmable microcomputer.

In the case of generating an arc profile, the swing center is fixed at the center of the arc. The truing tool swings around the center O while reciprocating vertically with a depth of cut Δ every stroke. Generally, a low grade GC grinding wheel is selected as the truing tool²⁾. The given depth of cut closely approximates the amount of GC wheel consumption δ , such that $\Delta - \delta$ is very small. The distance from the swing



(a) Transformation of cross-section



(b) Profile error

Fig.7 Arc truing result on CBN wheel

center to the GC wheel surface is therefore conserved. As result, the cross-sectional profile of the wheel is an arc resembling. If CNC equipment is used to control the swing center position, the above method can form a wheel of any desired convex profile.

Fig.7 shows the test result of the arc truing, starting with a straight vitrified bonded CBN280M200VB1 wheel, whose dimension are: 350mm diameter, 15mm width and 5mm diamond layer depth. Fig.7(a) demonstrates the transformation of trued wheel arc cross-section. The arc of cross-section took on the given swing radius of 30mm when the wheel has been entirely and uniformly trued. The tolerance in this case was less than $15\mu\text{m}$. Fig.7(b) shows the error of the profiled wheel. The deviation including the protrusion height of CBN particle was about $\pm 2\mu\text{m}$.

Fig.8 is the truing result in regard to an extremely fine grit diamond wheel: SD5000P150B7 with 300mm diameter and 10mm width, which is used to finish the groove of a ceramic bearing inner race. A family of soft (low grade) GC wheels in such order: #1500, #3000 and #6000, was used to perform from rough truing to precise (finish) truing. It is clear that the truing accuracy widthwise is within

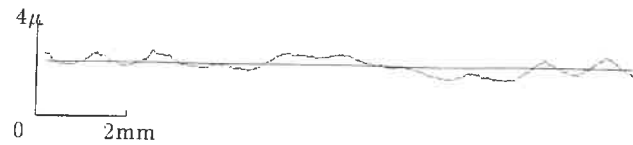


Fig.8 Arc truing result on diamond wheel

$\pm 0.5\mu\text{m}$. With the treated diamond wheel, a mirror surface of $R_{max}=50\mu\text{m}$ was obtained at infeed grinding a Si_3N_4 ceramic bearing.

4. CONCLUSION

In this paper a truing/dressing device was developed to enhance the total performance of superabrasive grinding wheel. Since the GC Cup-truer removes the wheel's bond material only, by "lapping" of the loose grains that are broken off from the GC wheel, it prevents abrasive particles from being damaged and allows to perform both truing and dressing simultaneously. The longitudinal infeed mechanism improves the truing accuracy, avoiding the influence of the tool wear on the generated profile. The experiment results demonstrated that the grinding performance of the diamond wheel trued by the GC Cup-truer is better than that of other truing tools, while the grinding performance of the CBN wheel is controllable by varying the grit size or the grade of the GC wheel applied. As one practical application of the GC Cup-truer, an Arc Truer was designed to produce a convex profile. An arc formation showed that the truing accuracy of the profile error below $1\mu\text{m}$ was obtained.

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INFLUENCE OF MICROSTRUCTURE OF WORK MATERIALS ON THE CUTTING TOOL WEAR

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1. Introduction

The sharpness of cutting edge is one of very important factor in the realization of precision machining. Therefore, the tool wear is especially serious obstacle to the precision machining.

In recent years, precision cutting of many kinds of materials (metals, plastics and ceramics) are expected. Therefore, it is very important that careful selection of tool material which is suitable for each work material.

Usually, work material hardness is regarded as a most important factor when selecting a tool material for reason of tool wear. However, this selective method is not always appropriate, because the tool wear does not always correspond to the work material hardness.

In this paper, the wear characteristics of nonmetallic hard tools (ceramics and CBN) in the cutting of high-carbon steel and of metallic tools (high-speed steel and cemented carbide) in the cutting of plastics (PVC) are considered, and the importance of microstructure of work material connected with tool wear is emphasized.

2. Experimental procedure

This cutting experiment is performed by the conventional turning of cylindrical work piece without coolant in lathe.

The metallic work materials are high-carbon steel (1%C, JIS SK3) and medium carbon steel (0.45%C, JIS S45C), and hardness of both steels are within from 200 to 800 Hv by heat treatment. The plastic work materials are PVC resins (four kinds of PVC-0, PVC-1, PVC-2 and PVC-3 according as content rate of inclusions).

The tool materials are alumina ceramic (Al_2O_3) and CBN for the cutting of both steels, and are high-speed steel (18-4-1 Co, JIS SKH4) and cemented carbide (K10) for the cutting of PVC resin.

Tool wear phenomena and the cause of the tool wear were investigated synthetically through the stresses and temperature on tool faces, elementary analyses of worn tool faces and microstructures of the work materials.

3. Experimental results and considerations

3.1. Wear characteristics of ceramic and CBN tools in the cutting of steels

Fig.1 shows the relation between work hardness (Hw) and tool flank wear (VB).

In the cutting of medium-carbon steel (S45C), VB of both tools decrease with the decrease in Hw. These phenomena are understood sensibly. On the contrary, in the cutting of high-carbon steel (SK3), VB of tools increase with the decrease in Hw within the range of low work hardness (less than 500 Hv). These phenomena are very remarkable and interesting fact.

The cause of these phenomena are considered as follows.

3.1.1. Consideration on the microstructure of the flank wear land

fine streaks scratched were observed all over flank wear land, but the adhesives were not observed.

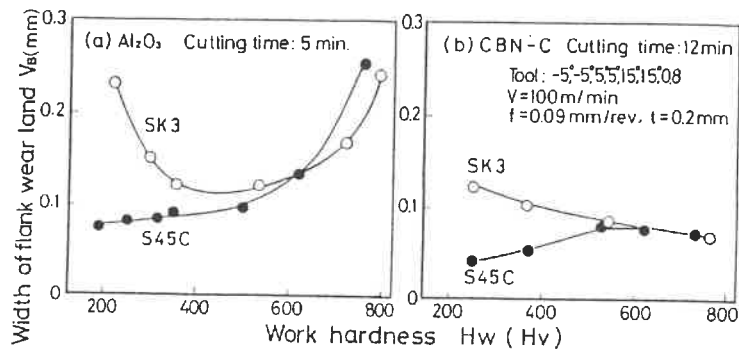


Fig. 1 Width of flank wear land vs. work hardness

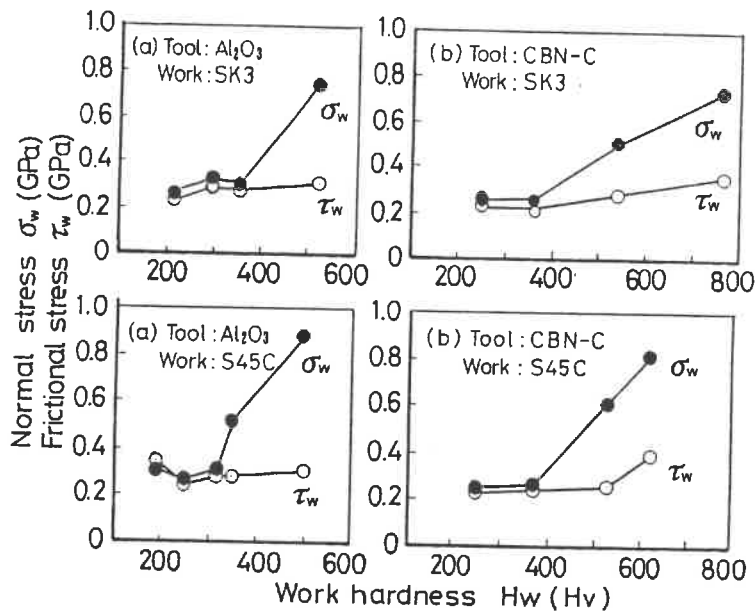


Fig. 2 Stresses on flank wear land vs. work hardness

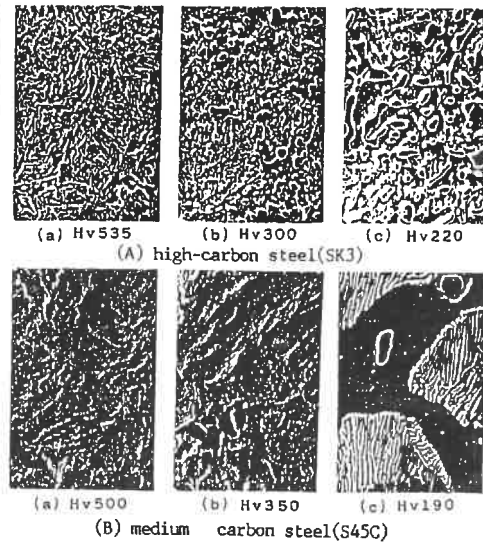


Fig. 3 Microstructures of work materials

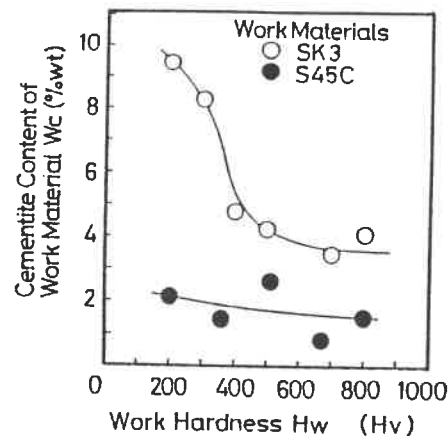


Fig. 4 Cementite content vs. work hardness

Therefore, it is expected that the flank wear land was received abrasive actions by hard particles in the work material.

3.1.2. Consideration on the stresses on the flank wear land

Frictional stress and normal stress on the flank wear land are directly and indirectly contributive factors to the increase of the flank wear, respectively. The stresses on flank wear land can not be obtained directly during cutting. But, these stresses can be obtained indirectly from the relation between area of the flank wear land and cutting forces.

Fig. 2 shows relation between work hardness and stresses on the flank wear land. Upper figures show the work material of high-carbon steel (SK3), and lower figures work material of medium-carbon steel (S45C).

In every case, normal stress (σ_w) increase with the increase in work hardness, but frictional stress (τ_w) do not increase in spite of the increase in work hardness. Therefore, the frictional stress has no strong influence on the change of the flank wear.

Moreover, in the cutting both work materials, the relation between the work hardness and the stresses are almost no different.

3.1.3. Consideration on the microstructure of work material

The change of work hardness is caused by the change of structure of work material by heat treatment. Therefore, the microstructure is important factor in the consideration on the tool wear.

Fig.3 shows microstructures correspond to hardness of work materials. Fig.3 (a),(b) and (c) are martensite, sorbite, and pearlite structure respectively. Upper and lower photos are the structures of SK3 and S45C respectively.

In the structures of SK3, there are relatively small cementite particles in sorbite structure, and there are great quantities of large cementite particle in pearlite structure. However, in the structures of S45C, these large cementite particles can not be found.

Fig.4 shows relation between the work hardness and cementite content rate (W_c) of the work materials. Cementite content rate of S45C remain unchanged in spite of the change of work hardness, and the content rate are very low.

However, cementite content rate of SK3 is fairly high ($W_c=9.5\%$) at 200 Hv of hardness, and it decrease clearly with the increase in the work hardness.

The cementite (Fe_3C) is very hard (1300 Hv), and these great quantities of cementite will influence strongly on the tool wear.

3.2. Wear characteristics of high-speed steel and cemented carbide tools in the cutting of PVC resin

Tool wear in the cutting of PVC resin is remarkable in spite of softness of PVC resin. Therefore, chemical properties and inclusions of PVC resin are regarded as the influential factors on the tool wear.

3.2.1. Consideration on the influence of inclusions of PVC resin on the tool wear

Content rate of INclusions of PVC resins (four kinds of PVC-0~PVC-3) were measured as residua by combustion method and Soxhlet extraction method (solvent extraction method). The elements contained in the residua were identified by spectroscopic analysis.

Fig.5 shows relation between rate of residua by combustion method and it

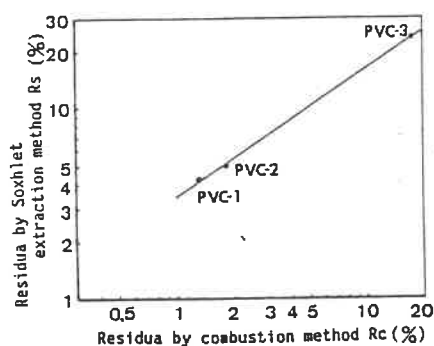
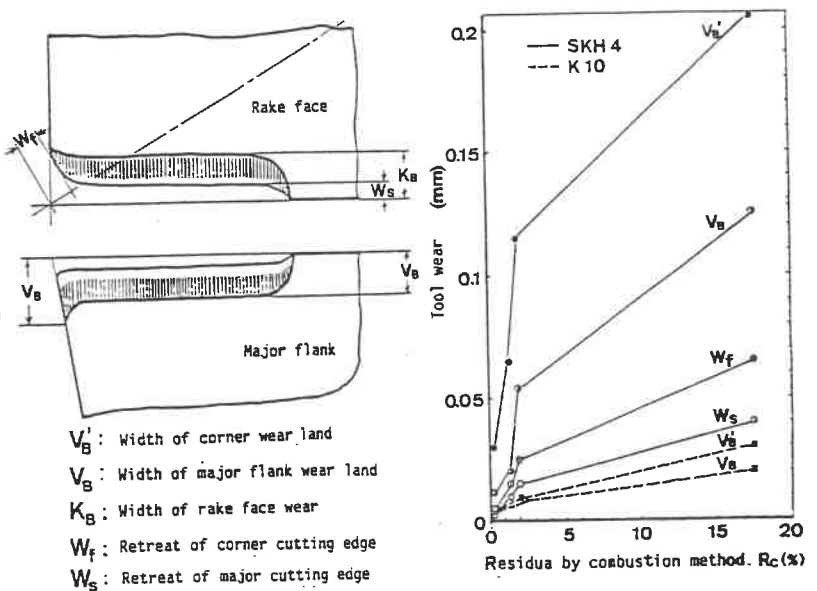


Fig.5 Residua by Soxhlet extraction method vs.it by combustion method

Table 1 Elements in residua by combustion method

	Ca	Pb	Al	Si	Ti	Fe	Cu	Mg	Sn
PVC-0	2	1	2	2	1	2	3	2	3
PVC-1	2	4	3	3	3	2	2	2	3
PVC-2	3	4	3	3	3	3	2	2	T
PVC-3	5	3	2	2	2	2	1	2	-

note: $5 > 4 > 3 > 2 > 1 > T$



Shape of tool (0,10,10,10,30,30,0) $V=100\text{m/min}$ $t=0.5\text{mm}$ $S=0.06\text{mm/rev}$ $T_c=10\text{min}$

Fig. 6 Tool wear vs. residua by combustion method

by Soxhlet extraction method.

Table 1 shows result of qualitative analysis of elements in the residua by combustion method. these elements are contained in PVC resin as CaCO_3 , PbSO_4 , Al_2O_3 , SiO_2 , TiO_2 , Fe_2O_3 , MgO and so on.

Fig.6 shows relation between residua content rate(R_c) and tool wear.

The tool wear of high-speed steel increase quickly with slight increase in the content rate of residua. This phenomenon depend on the formation of reaction products(ex. FeCl_2) and the removal of them by abrasive actions of the inclusions in the PVC resin.

3.2.2. Consideration on the microstructure of worn tool face

Fig.7 shows the microstructures of worn tool face of high-speed steel and cemented carbide after cutting of PVC-3.

Both tool faces become rough as like ascorroded surface. Temperature of cutting edge under this condition was about 400K.

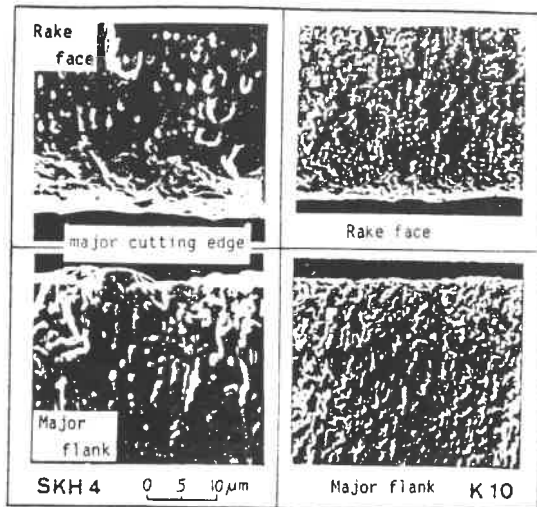
3.2.3. Consideration on the chloride

Fig.8 shows the result of EPMA line analysis of Cl on the cross section of worn iron tool. The peak of X-ray intensity curve on Cl appear keenly at tool face, it is confirmed the formation of chloride(FeCl_2 or FeCl_3) on tool face.

4. Conclusions

The main results of these investigations are as follows:

- [1] In case of the cutting of high-carbon steel by ceramic and CBN tools
 - (1) The flank wear both tools increases with the decrease in work hardness.
 - (2) The increase in flank wear with the decrease in work hardness is due to reinforcement of abrasive action of cementite(Fe_3C) in the work material.
 - (3) The microstructure of work material exert great influence on the flank wear.
- [2] In case of the cutting of PVC resin by high-speed steel and cemented carbide tools
 - (1) Hard particles are contained in PVC resin as additives, and these particles scratch the tool face.
 - (2) Tool wear increase with the increase in quantity of these particles in the PVC resin.
 - (3) At high cutting temperature, degradation of PVC resin chemical reaction products are generated on the tool face, and tool wear increases quickly.



Shape of tool(0,10,10,10,30,30,0) S=0.06mm/rev t=0.5mm

Fig.7 Microstructures worn tool faces after cutting of PVC-3($v=100$ m/min)

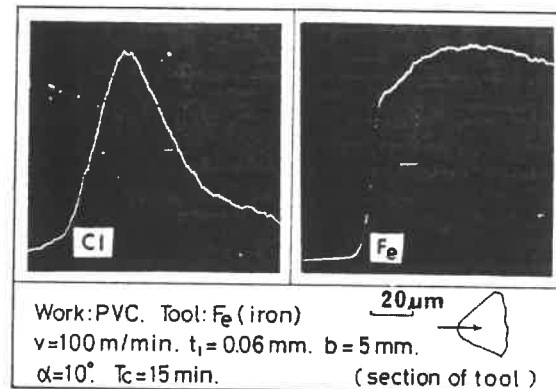


Fig.8 EPMA line analysis of Cl on the cross section of worn iron tool

Performance of High Flux Heat Exchangers Fabricated by Diamond Machining

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A traditional compact heat exchanger has been defined as a device with a surface area density (active surface area per unit volume) of between $700 \text{ m}^2/\text{m}^3$ and $5,000 \text{ m}^2/\text{m}^3$ {1}. A typical automobile radiator has a surface area density of 1,000 whereas the human lungs have a surface area density of over 17,000. In heat exchangers, the upper limit came about as a result of conventional manufacturing methods. However, heat exchangers with a surface area density over $5,000 \text{ m}^2/\text{m}^3$ have been built and tested. The use of diamond machining methods allows for this increase in performance. Heat exchangers with a surface area density over 5,000 have been defined as micro compact heat exchangers because the hydraulic diameter of the fluid flow passages must be in the micrometer range to attain the high density. In addition to the large surface area density, the conduction resistance between the two fluids may be reduced to tens-of-micrometers of high conductivity copper.

The performance of micro compact heat exchangers has been modeled and compared with experimentation. The performance model is an iterative approach whereby the input is the required total heat transfer capacity, the mass flow rates of the hot and cold fluids, and the allowable core pressure drop. The free flow area for each fluid and the overall core dimensions are determined by the allowable pressure drop. There is an assumption as to the mass velocity on one side of the heat exchanger (cold fluid). The NTU-effectiveness method is then used to determine if a heat balance exists. If not, another value for the cold-side mass velocity is assumed and the process repeated until a heat balance is achieved. The complete approach may be found in reference {2}.

The actual performance of a micro compact heat exchanger was determined experimentally. The heat exchanger plates were cut on an air spindle with a trapezoidal, single crystal diamond tool. The tool geometry and machining setup is shown in Figure 1. The plate material was UNS-C10200 copper foil 125 micrometers thick and 25mm wide. The foil was scored in a cross-cut fashion to a depth of about 100 micrometers. The channels were then machined to a depth of 80 micrometers. This provided a method for separating the square foil coupons without destruction of

the flow channels. The individual foils were etched in 1:1 nitric acid for 20 seconds. The clean coupons were stacked so the channels in one foil were at a right angle with the channels in the adjacent coupons. The stack was clamped with a compressive stress of 60% of material yield and placed in an oven at 600C and 10 microTorr for 30 minutes for bonding. The process resulted in a cross-flow micro compact heat exchanger with a surface area density of $6,876 \text{ m}^2/\text{m}^3$. A typical configuration is shown in Figure 2.

The heat exchanger was placed in a test loop and tested under a range of flow conditions. The hot-side inlet temperature was maintained in the range from 40C to 76C while the cold side was maintained between 22C and 23C by regulating the mass flow rate of both fluids equally. The core pressure drop for the hot side ranged from 0.5 bar up to 2.3 bar while the drop on the cold side ranged from 0.8 bar to 3.5 bar. Flow on both sides was maintained in the fully developed laminar region with a Reynold's number between 72 and 492. The mass flow rates of water were equal for both sides and varied between 14 grams/second and 45 grams/second. Because of a less than desirable bond between the heat exchanger core and the steel manifold housing (resulting in approximately 0.0007% leakage), the device was operated under conservative conditions of relatively low pressure and temperature. Although this is a very small leak rate, it would not be tolerable in certain applications such as in a vacuum or heat transfer between non-compatible fluids.

The performance of the heat exchanger is shown in Figure 3. For the test conditions, the maximum overall heat transfer coefficient was $6.4 \text{ KW}/\text{m}^2\cdot\text{K}$ and the volumetric heat transfer coefficient was $44 \text{ MW}/\text{m}^3\cdot\text{K}$ of log-mean temperature difference. Comments on this performance will be made subsequently. The actual effectiveness of the heat exchanger was determined from the inlet and outlet temperatures. The effectiveness may be viewed as the efficiency of the device based on the absolute maximum heat which could be transferred based on temperatures only with no losses or resistance. The comparison between the effectiveness as predicted by the model and that actually measured is shown in Figure 4. The maximum effectiveness was measured to be 43% whereas the model predicted 48%.

Conclusions

A model, a fabrication process, and a series of performance tests were conducted for a typical micro compact heat exchanger. Under conservative operating conditions the device exhibited a very high thermal capacity. The trapezoidal-shaped channels can be replaced by rectangular channels in future designs. The performance model predicts a volumetric heat transfer coefficient over 7 times that of the present design for the same overall volume. Such devices can find application where many kiloWatts of heat must be transferred in a small volume and small weight. Although it is not being

suggested here, a device operating with the same fluid temperatures as are found in an automobile radiator (a temperature difference of 67C), could perform the same thermal task as a conventional radiator with a greatly reduced volume and weight. For a 150 horsepower engine rejecting 1/3 of the power to the cooling system, the micro compact heat exchanger tested would require an active volume of approximately 13 cubic centimeters (0.77 cubic inches) and a weight of less than 1.1 Newtons (4 ounces). Again, such an application is not practical for these devices, but it does illustrate their tremendous thermal capacity.

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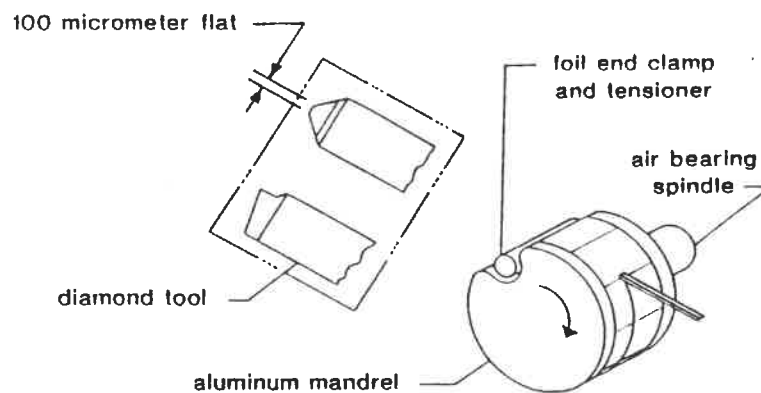


Figure 1. Setup for diamond machining micro channels

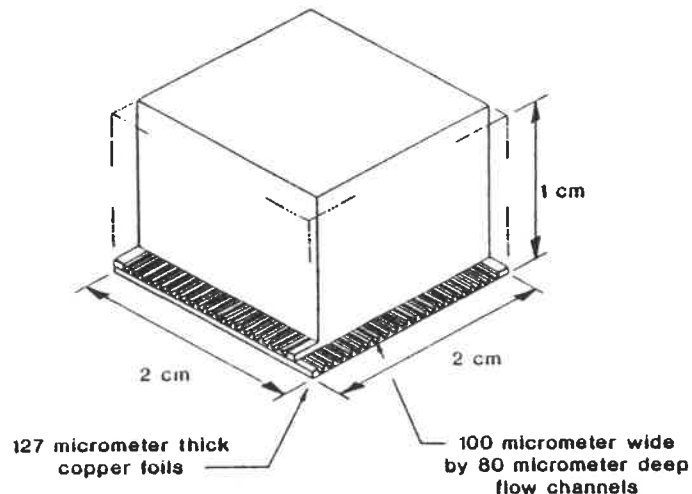


Figure 2. Typical micro compact heat exchanger

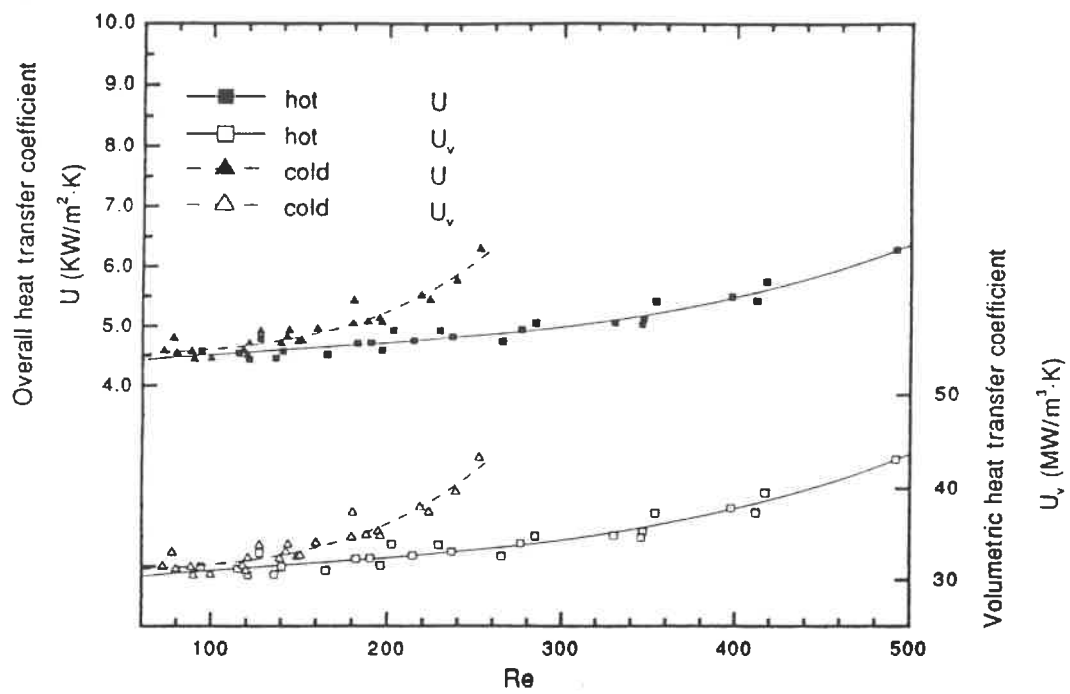


Figure 3. Micro heat exchanger thermal performance

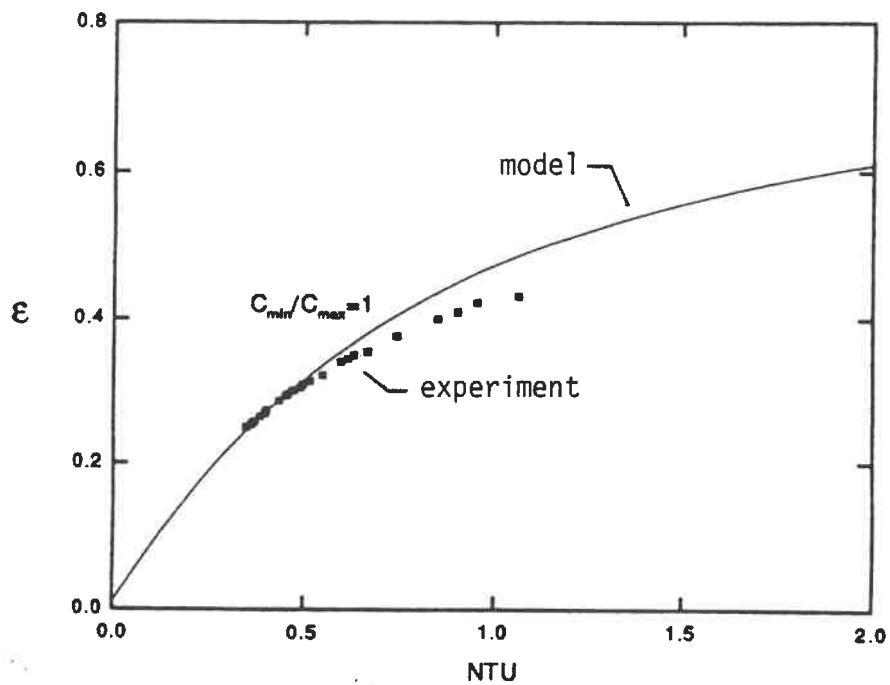


Figure 4. Micro heat exchanger effectiveness

Detection of Acoustic Emission Signal from a Rotating Cutting Tool

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ABSTRACT

This paper presents a monitoring method of cutting process by using acoustic emission (AE) signal sampled during cutting through a rotating cutting tool. The AE signal is transmitted through cutting fluids. The adaptive noise canceler is applied to get rid of mechanical noise. The experimental results show the high reliability of the proposed method.

1. INTRODUCTION

Monitoring of the cutting process is one of the most important technologies for improving the reliability of unmanned manufacturing system. In recent years, application of acoustic emission (AE) sensor for monitoring the cutting process has attracted the interest. Since the AE signal is greatly influenced by the path which the signal is transmitted, the AE sensor should be fixed by keeping the distance constant between the sensor and the cutting point. To meet this demand, for the application of AE sensor to the machine tool with a rotating cutting tool, the following two problems should be solved:

First, the AE sensor should be attached to the rotating cutting tool side to keep the distance constant. In addition, the sensor and wires to transmit the signal should not interfere the generating motion of the machine tool. In this study, we proposed to use cutting fluid as media to transmit the AE signal. With this proposed method, it became possible to pick up the AE signal from a rotating cutting tool keeping the distance between the sensor and the cutting point constant. Needless to say, the signal gain is affected a lot by the condition of fluid flow. The frequency response measurement of the AE signal in the fluid flow was, therefore, carried out to make clear the propagation characteristics.

The second problem to be solved is to get rid of the noise generated from the bearing system. Besides, the frequency characteristic of the mechanical noise changes greatly with the change of rotating speed of the spindle. A filter to get rid of the mechanical noise should be, therefore, adapted to the change of the spindle rotating speed. In this study, adaptive noise canceler is applied to signal treatment and an effective method to obtain optimal parameters in the adaptive filter is proposed.

2. FREQUENCY RESPONSE MEASUREMENT

In order to measure the transfer function of the AE signal in fluids, the test devices as shown in Fig.1 were used. Fig.1 Model B shows the test device which imitates the cutting fluid supply system. Fig.2 shows the system for analysis. Fig.3 shows the transfer function obtained with the Model A, while Fig.4 shows the case of Model B which contains the noise generated through the collision between fluids and solid in the frequency range lower than 100kHz. Since the gain in the frequency range from 100kHz to 400kHz changes markedly depending on the frequency, the frequency range higher than 400kHz is recommended to use for signal detection. Compared with the case of Model A [Fig.3], the transfer function of Model B is much more influenced by the flow rate as shown in Fig.5.

3. ADAPTIVE NOISE CANCELER (ANC)

The structure of ANC is illustrated in Fig.6. ANC requires two initial parameters M and μ to be set in its adaptive filter. M is a number of weights for adjustment and μ is a rate of adjustment [1]. These parameters affect the filtering performance which is generally represented by the learning curve as shown in Fig.7. In this study, the filtering performance is evaluated by following two parameters:

- (1) Error of filtering $\text{Min}.V(n)$, which is used for evaluating the filtering accuracy.
- (2) *Inclination* of the learning curve in $n\text{-log}V(n)$ space which is used for evaluating the convergence rate.

When M or μ is small, both the $\text{min}.V(n)$ and *Inclination* are small. Since initial parameters interactively influence the filtering performance, it is difficult to obtain their optimal combination. To get optimal combination, therefore, a new parameter is proposed.

In order to converge the learning curve, the parameter μ must satisfy the following condition.

$$\mu \leq \frac{2.0}{M\sigma^2}$$

where σ^2 is the variance of input [2]. In this study, μ is given as:

$$\mu = C_{\text{ANC}} \frac{2.0}{M\sigma^2}$$

where C_{ANC} is a new parameter and has the value from 0 to 1. Accordingly, μ is decided by setting M and C_{ANC} . M dominates the reliability of adjustment and the computer calculation time. The larger the value of M , the higher the reliability of the filter. In that case, however, the calculation time to be needed becomes long. For the on-line monitoring system, the calculation time must be shorter than the data sampling interval. Therefore, M should be largest while satisfying the time restriction. The influence of C_{ANC} to the filtering performance is independent to M , and its relation is shown in Fig.8. Hence, C_{ANC} can be determined considering the requirement for the filtering performance.

4. EXPERIMENTAL RESULTS

In order to confirm the effectiveness of the proposed method, detection of cutting edge breakage is carried out by the experimental set up shown in Fig.9. Both the primary input and the reference input are detected through cutting fluids. Fig.10(a) shows the frequency characteristics when a normal tool was used, while Fig.10(b) shows the case of broken tool. The frequency characteristic in the case of broken tool contains characteristic peaks which are not visible as far as the normal tool is used [3].

5. CONCLUSIONS

[1] A new method to detect the AE signal from a rotating cutting tool is proposed. Cutting fluids are used as media to transmit the AE signal.

[2] ANC is applied to get rid of the mechanical noise generated from the bearing system. The method to obtain the optimal parameters in the ANC is proposed.

[3] The effectiveness of proposed method is confirmed through the detection of cutting edge breakage in the milling process.

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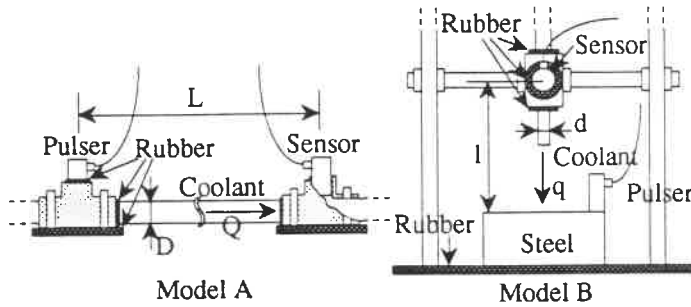


Fig. 1 Experimental devices

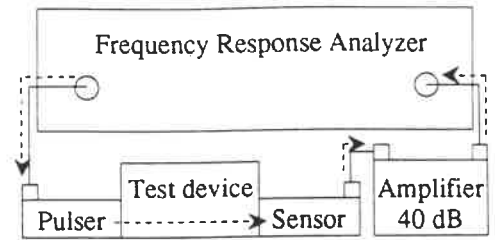


Fig. 2 Equipment for frequency response measurement

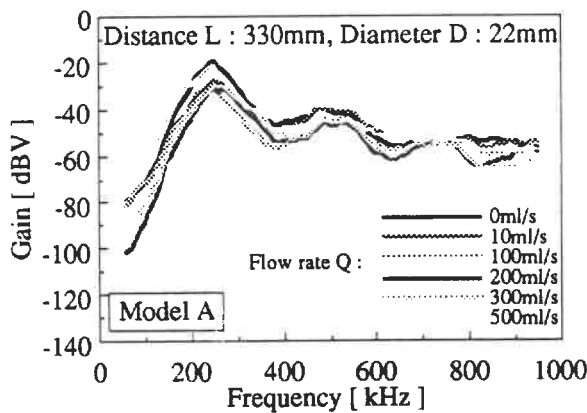


Fig. 3 Transfer function - 1

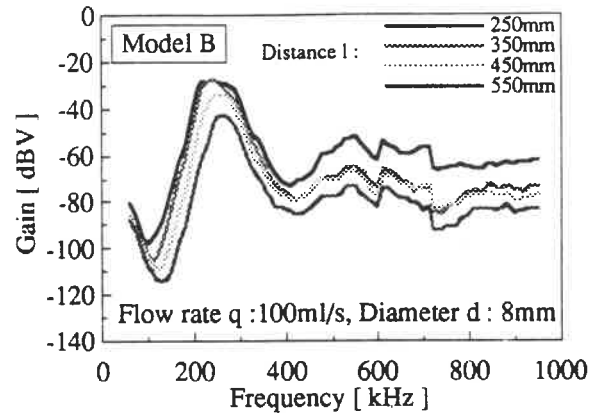


Fig. 4 Transfer function - 2

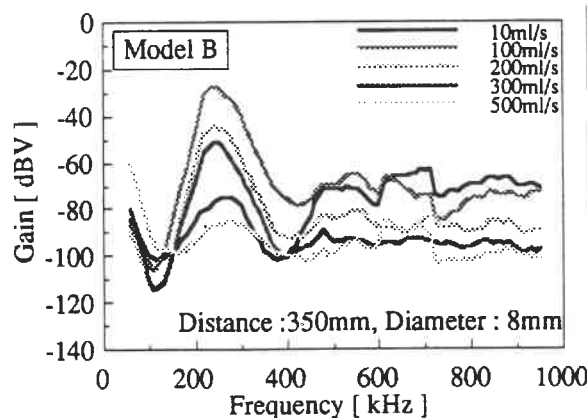


Fig. 5 Transfer function - 3

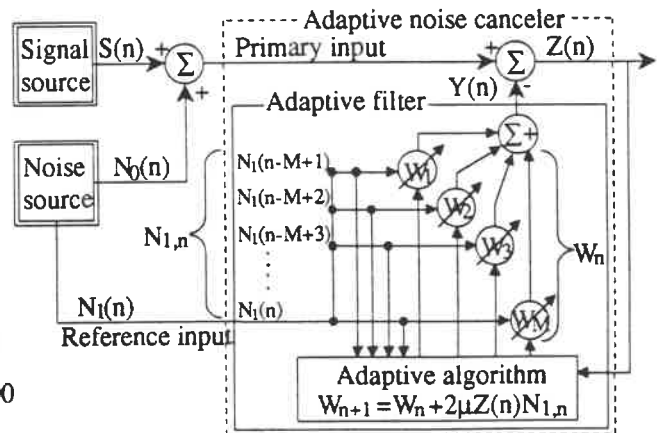
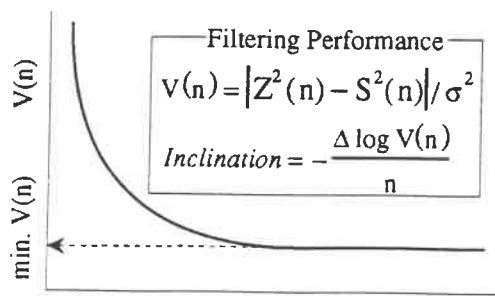


Fig. 6 Structure of adaptive noise canceler



Number of adaptations n
Fig. 7 Learning curve

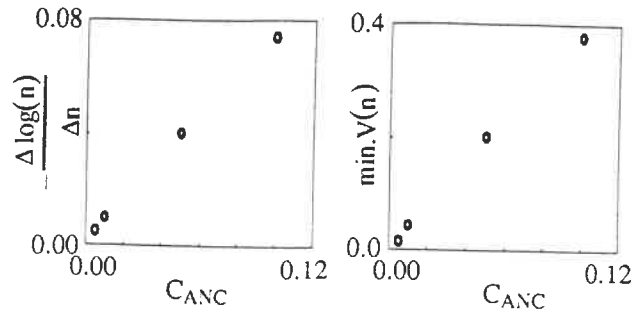


Fig. 8 Relationship between C_{ANC} and the filtering performance

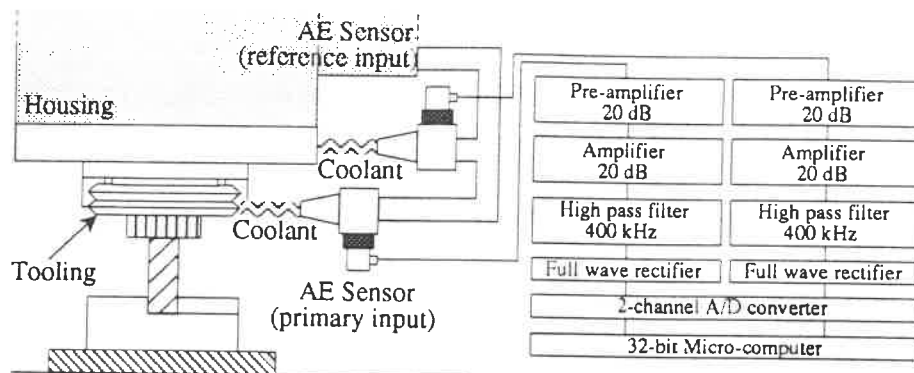


Fig. 9 Experimental set up

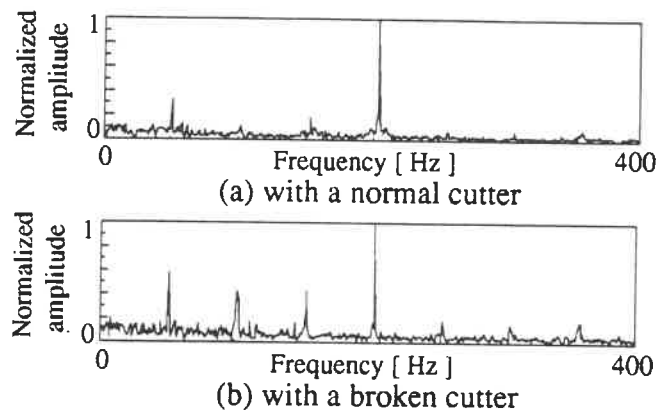


Fig. 10 Detection of cutting edge breakage
rotational frequency 3000rpm, radial depth 0.1mm, axial depth 2mm, feed rate 200mm/s, $M=16$, $C_{ANC}=0.01$
square-endmilling cutter with 4 cutting edges

APPLICATION OF A MODIFIED BACK-PROPAGATION ALGORITHM FOR PATTERN CLASSIFICATION IN TURNING

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1.Introduction

In this paper a distributed neural network has been applied to a pattern recognition problem for classification of tool wear in turning, to discriminate between a worn-out tool and a fresh tool. Tool wear can be monitored by measuring cutting forces, vibration, acoustic emission etc. In addition to measurement of these parameters, strategies are required to control the machining online automatically. This can be implemented with rule based systems such as statistical techniques (Group method data handling, Multiple regression analysis) and Artificial Neural Network (ANN). The advantages of ANN over statistical methods are quick information processing in a distributed and parallel manner, adaptable to change in environment, quick-decision making and online updation of information from the environment.

1.1 Experimental work

Experiments were carried out on a precision high speed VDF lathe. The turning operation was carried out on a spheroidal graphite cast iron (SGCI). Data collected during machining SGCI include static forces - axial force (F_x), radial force (F_y), tangential force (F_z) and flank wear land width of tool (V_b), center line average (R_a) of the workpiece, maximum profile depth (R_t) of the workpiece. Static forces and flank wear measurements were made at different intervals of time. Depending on length of cut, machining was stopped after 60-80 seconds to measure V_b , R_a and R_t . Static forces

were recorded at 2 to 3 intermediate points. The set of measurements immediately prior to a wear measurement was taken for training the ANN. Patterns with maximum variance have been used for training the ANN.

2.Back-propagation Algorithm

The back-propagation algorithm uses an objective function, which is defined to be the summation of squared errors between the desired outputs and the network outputs. It then employs the steepest-descent search algorithm to seek the minimum of the objective function. The algorithm works on supervised learning. The algorithm can be implemented with a sequence of perceptrons. The state of a perceptron is the summation of inputs and weights plus threshold. The output of the perceptron is a non-linear function (sigmoid) which is continuously differentiable. The output of a neuron is the input to the adjacent neuron. The sigmoid function describes the input-output characteristics of each neuron. The objective function is minimized by adjusting the weights and thresholds by back propagating the error.

2.1 Disadvantages of Back-Propagation Algorithm

The number of cycles required for the objective function to reach the desired minimum is very large. The objective function does not reach the desired minimum due to some local minima whose domains of attraction are as large as that for the global minimum. The algorithm converges to one of those local minima and hence learning stops prematurely or the value diverges. The updating of weights will not stop unless every input is outside the significant update region (0.1 to 0.9) and the outputs of the network will be approaching either 0 or 1. This requires thousands of iterations for the convergence of the algorithm.

3.Results

3.1 Conventional Back-Propagation Algorithm

Data obtained during machining were normalized between 0 and 1. Training of the network was done using 6 neurons in the input layer and 3 neurons in the output layer. Neurons in the hidden layer was varied between 5 and 20 with learning factor as 0.5. The final configuration of the network is 6-15-3 with the objective function reaching 0.0011 for 20000 cycles. When new patterns were tested the matching was 95%.

3.2 Selective Update Back-Propagation Algorithm

The data collected during machining were binary coded. Two selection criteria were used in the Selective update Back-propagation algorithm. In the first criteria, the input is classified if $D < 0.5$ and misclassified if $D \geq 0.5$, where

$$D = | \text{target output} - \text{network output} |$$

In the second criteria the input is checked if it is in the insignificant update region when

$$1 - \epsilon > D \geq 0.5$$

Case 1: without hidden layer, inputs in the misclassified sets are updated with learning factor of 0.05. The final mean squared error was 0.4 and number of cycles reached was 100. When new patterns were presented matching was 100%. In case 2, hidden layer was used. When the inputs in the misclassified sets are empty updating is stopped. Training was done with a learning factor of 0.05 and with one hidden layer. The final mean squared error was 0.5 with 1000 iterations.

3.3 Algorithm used for Selective Update BPA

Step 1: Randomly initialize the weights

Step 2: Present the inputs and desired outputs.

Step 3: Compute network output 'x' by

$$x = 1 / [1 + \exp[-(\sum_{j=1}^n w_j x_j + t)]]$$

Step 4: Determine V^m , the set of valid update data by

$$1 - e^{-D} \geq 0.5$$

if V^m is empty do not compute weights. Goto step 8.

Step 5: Compute the objective function by using

$$J = \frac{1}{2} \sum_{x \in V^m} [\text{Desired output} - \text{Network output}]^2$$

Step 6: Adapt weights and thresholds by the following eqn.

$$w(n+1) = w(n) + \eta \delta$$

Step 7: Repeat by going to step 3.

Step 8: Change the sigmoid function of the output neuron to the signum function.

4. Conclusions

Application of neural network with supervised learning for tool wear monitoring has been discussed. Performance of the conventional Back-Propagation Algorithm and its performance when selection criterion is used with hidden layer and without hidden layer have been discussed. When selection Criterion are used training is fast and pattern classification is 100 %.

EVALUATION OF ENVIRONMENTALLY SAFE CLEANING AGENTS FOR DIAMOND-TURNED OPTICS

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Traditionally, machining fluids, mineral oil in this case, were cleaned from diamond-machined metal optics by flushing with Trichloroethane (TCA) and then rinsing with alcohol. TCA, however, was ruled environmentally unsafe and will be phased out of use by 1995.

This project involves an effort to find a suitable replacement for TCA in the cleaning of diamond-machined optics. The cleaning ability of eleven solvents and seven detergents was tested and compared against the standard, TCA. All test cleaners were environmentally safe and commercially available. Oxygen Free High Conductivity Copper was selected as the test substrate because it is commonly used in metal optics applications.

Ninety-six copper specimens were diamond-machined and immersed in mineral oil for subsequent cleaning. The cleaning procedure involved randomly selecting a specimen for test, immersing it in a bath of test cleaner, and ultrasonically cleaning it for five minutes. Those specimens cleaned with solvent (including the standard - TCA) were immediately immersed in ethanol and ultrasonically rinsed for five minutes following the solvent cleaning. The specimens cleaned with detergent were immersed in deionized water, ultrasonically rinsed for five minutes, and then flushed in a stream of deionized water after the detergent cleaning. Finally all of the specimens were blown dry using clean nitrogen.

Surface film thickness proved to be a good criterion for determining surface cleanliness and subsequently the effectiveness of the cleaners. Ellipsometry was employed to determine the residual film thickness after cleaning the test specimens because it produces quantitative results with submonolayer sensitivity.

Ellipsometry measures the state of polarization of an elliptically polarized light beam. This state of polarization changes abruptly as it is reflected or refracted at the interface of two optically dissimilar media. This change is measured in terms of two parameters, the amplitude ratio (Ψ) and the phase difference (Δ).

Computer software has been developed to model this system, calculating film thickness in terms of Ψ and Δ . The model requires input of the complex refractive index of the substrate, the refractive index of the film layers, the angle of incidence of the light beam, and the wavelength of the incident light.

Before the surface film thickness could be determined, it was necessary to determine the composition of the surface layer. Since bare copper surfaces readily oxidize in air, the surface layer could be composed of either copper oxide or a solvent and mineral oil mixture. A deuteron probe analysis was conducted as well as a preliminary ellipsometric analysis, both indicating that the surface film was composed only of a solvent and mineral oil mixture with no oxide present.

Ψ and Δ were calculated from the angular settings (in degrees) that were recorded at the two points of extinction of the signal. They were measured at three points on the surface of each specimen. These were then averaged to provide one value of film thickness for each cleaner at a given date. Solvent tests were conducted in four replications of twelve specimens each. Detergent tests were conducted in six replications of eight specimens each.

An ANOVA (analysis of variance) was performed to determine the significance of the differences in the cleaning abilities of the test cleaners. The ANOVA indicated a significant difference in cleaning abilities of the solvents, but little difference in the cleaning abilities of the detergents. These results were substantiated using a Least Significant Difference (LSD) test which provides a comparison of the different solvents. This test indicated that over 95% of the time the cleaner that produced the best results, X-Caliber, would leave a thinner residual film than that left by TCA..

Figure 1 compares the residual film thickness after cleaning with each of the test cleaners. The data represent average film thickness for all replications using that cleaner. It can be seen that 11 cleaners perform as well or better than TCA. One of these should be adaptable to nearly every application. Cost is also a consideration in selecting a cleaner. Figure 2 provides a cost comparison of all of the test cleaners. Taking both cleaning ability and cost into consideration, Isopar M appears to be the best choice of a cleaner for diamond-machined copper optics.

Residual Film after Cleaning

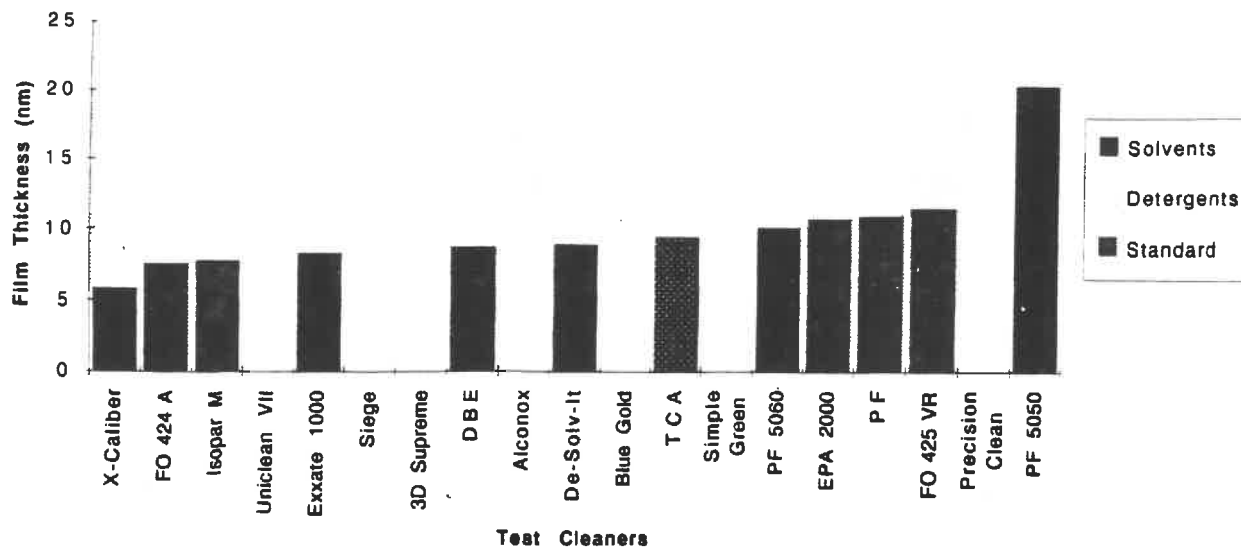
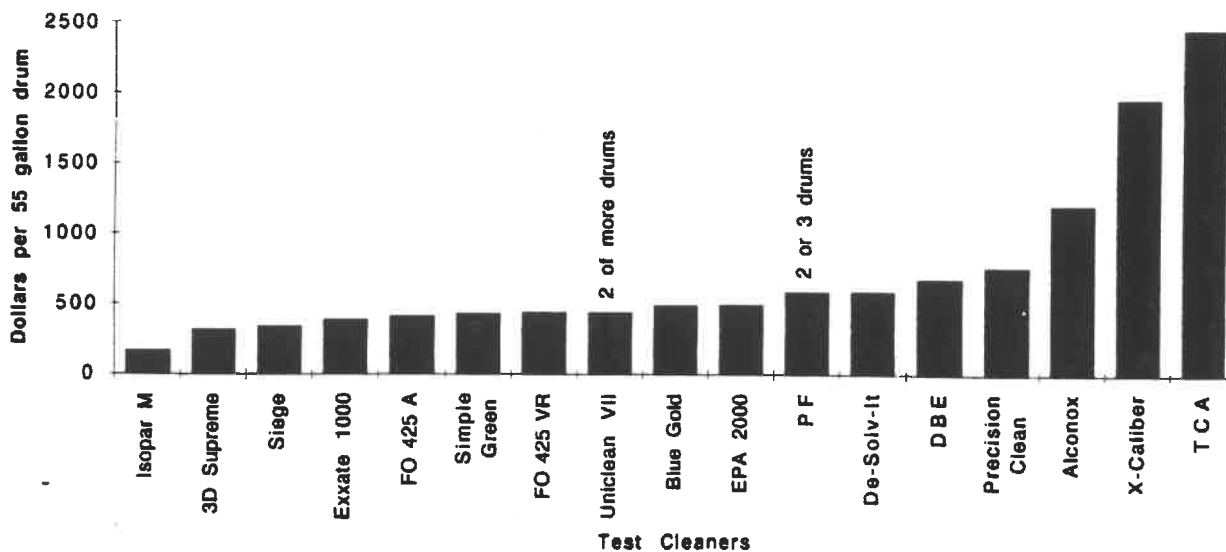


Figure 1

Comparative Cost of Cleaners



Cost of PF5050 and PF5060 were omitted because at \$9000/drum they distorted the scale.

Figure 2

The Influence of Crystallographic Orientation on Wear Property of Single Crystal Diamond Cutting Tool

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A single-crystal diamond has a large anisotropy in lapping and frictional characteristics, including the coefficient of friction and lapping stock removal depending on its crystallographic orientation. From these facts, it might be assumed that in diamond turning using single-crystal diamond tools (hereafter "diamond tools"), the cutting behaviors would largely depend on the plane of the diamond crystal (the rake face) on which the chip is directed, and the direction the chip flows. If the crystallographic orientations of the rake face and tool edge are determined, the crystallographic orientations of the flank should be determined according to the tool shape. Attention should be paid to the crystallographic orientation of the flank and cutting edge which might have affects on the finished surface quality, and tool wear. Moreover, the rough diamonds have the individual difference of lapping difficulty each other, even on the same plane and direction. Therefore, the wear performance of diamond tools are effected by the crystallographic orientations and wear characteristics of the rough diamond.

The effects of diamond anisotropy on tool wear were examined, using diamond tools of four kinds of crystallographic orientation. And, these four diamond tools were lapped up from only one dodecahedral rough diamond, eliminating the differences of individual characters of rough diamond used as tool material. Figure 1 shows the sawing method of diamond that is divided into four kind of their crystallographic orientations.

The diamond tools prepared are the circular arc tools having rake angle of 0° , a clearance angle of 5° , and corner radius of 1 mm. The crystallographic orientations of tools were aimed at as follows: plane $\langle 100 \rangle$ for rake face and direction $\langle 100 \rangle$ for tool edge direction, $\langle 100 \rangle / \langle 110 \rangle$, $\langle 110 \rangle / \langle 100 \rangle$ and $\langle 110 \rangle / \langle 110 \rangle$. These planes of $\langle 100 \rangle$ and $\langle 110 \rangle$ are effectively easy lapping planes. And the two directions of $\langle 100 \rangle$ and $\langle 110 \rangle$ on their planes are direction of the minimum and maximum values for the coefficient of friction, and direction of the most easy and difficult lapping, respectively.

The finished tools (tool No. T411 ~ T422) were inspected for the crystallographic orientation with X-ray Laue back diffraction method. The finished tools, however, did not slightly meet the desired

specifications in crystal plane and direction, due to errors during tool making. The desired and actual values of crystallographic orientation of these tools are shown in Fig. 3.

Figure 2(a)~(f) show the drawing method of Fig 3. Figure. 2(a) is the reference sphere of diamond. The unit spherical triangle is projected with Mercator's method positioned 001 point as the north pole. This area is shown in Fig. 2(b). In this Mercator's projection area, the flow direction of chip on rake face and rubbing direction on finished surface are possible to be represented without the deviation of direction. Fig. 2(c), (d) shows the relationship between corner radius r_ϵ and reference sphere at tool edge. In tool edge, a cylinder having the radius r_ϵ is hypothesized, and whose side face is corresponded to flank of tool edge portion. The inscribed reference sphere in this cylinder and the tool rake face are shown in Fig. 2(d). Cross line between the reference sphere and the flank in cutting area coincides to a group of crystal planes of flank. The group of crystal planes concerned in cutting are shown as fan-shaped zone on the reference sphere in Fig. 2(e). Fig. 2(f) shows the spread chart of the fan-shaped zone by combinations of the unit spherical triangles.

Figure 3 shows the route diagrams of crystallographic orientation of four tools from face to flank. The chart is consisted of the 12 parts. Each parts are combined with 4 unit spherical triangle segments which have the area enclosing by the representative three planes: plane (100), (110) and (111) in (the reference sphere of) diamond crystal. These unit triangle spheres were represented by the Mercator's projection method, not but by the stereographic projection method.

Using these tools, cutting experiments were carried out on following cutting conditions: work material, Si hyper-eutectic Al alloy (cylinder of diameter ϕ 70 mm with hole ϕ 15 mm, length 200 mm): cutting speed, 4.6→1.6 m/s (face cutting on end of cylinder from outside to

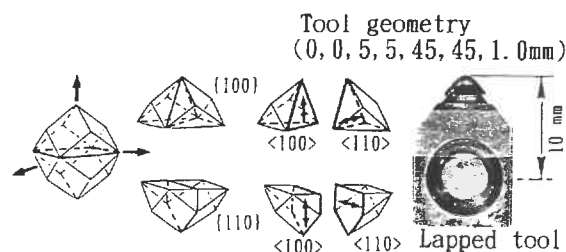


Fig. 1 Method of slices of diamond

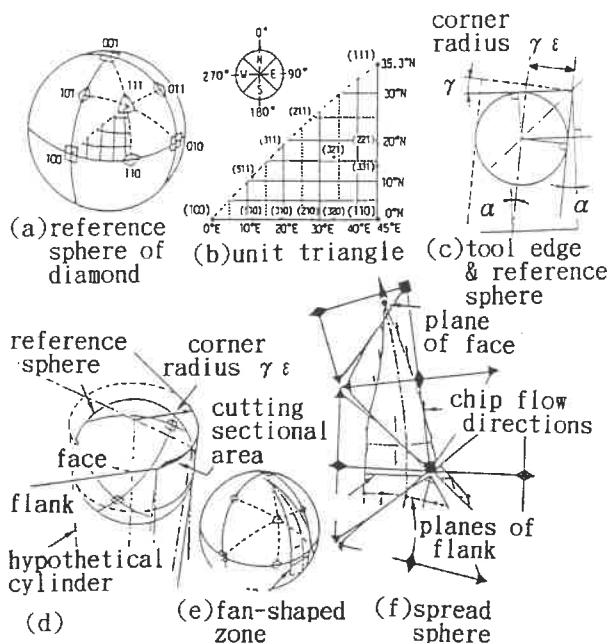


Fig. 2 Drawing method of crystallographic orientation routes of tool edges by restricted Mercator's projection

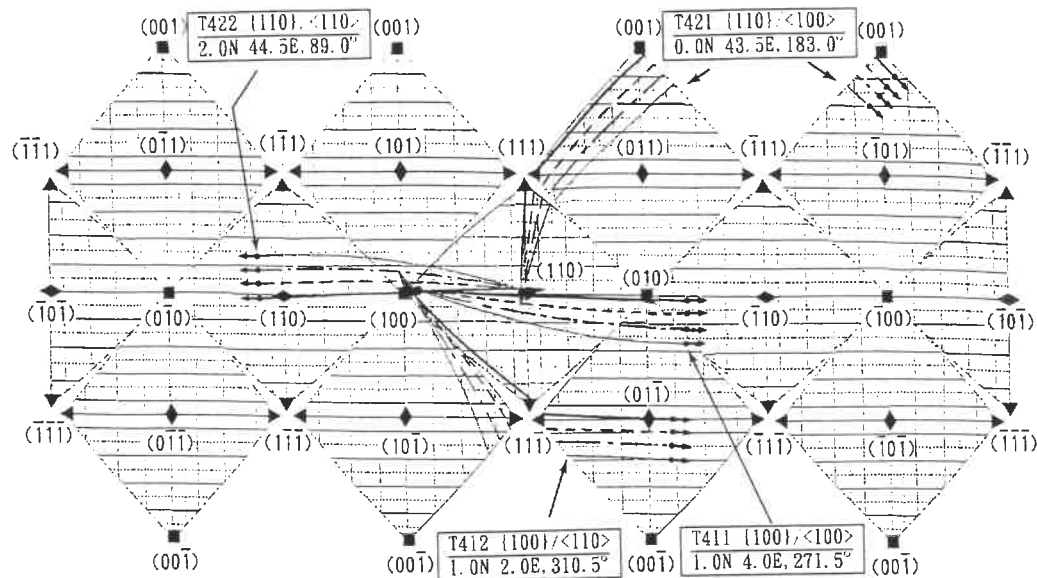


Fig. 3 Diagram of crystallographic orientation route of tool edges on reference sphere by the restricted Mercator's projection

center): depth of cut, $40 \mu\text{m}$; feed, $40 \mu\text{m/rev}$; dry cutting.

Figure 4 shows the microphotographs of worn tools after 5 km cutting distances. The different tool wears were occurred each other. By Electron Probe Surface Roughness Analyzer (SEM with 4 detectors, ELIONIX MULTI-ch-SEM ERA-8000), the topographies of worn tool edges are examined. Part of the topographies are shown in Fig. 5. The observed point of tool edges shown in Fig. 5 are corresponded to the positions

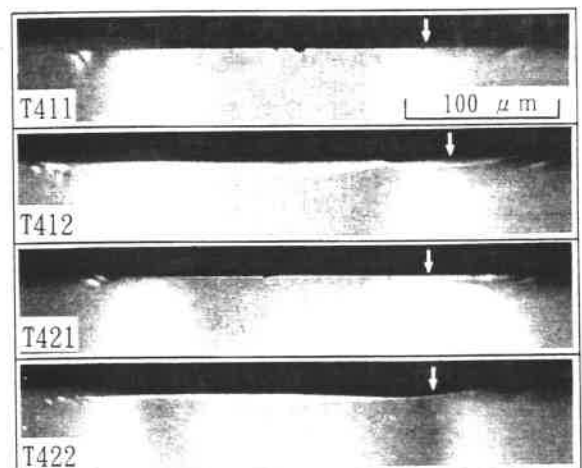


Fig. 4 Microphotographs of tool wear

with arrow head in Fig. 4 respectively. Figure 5(a) shows the contour line of worn tool edge. The relationships between the wear amount and the position on route of crystallographic orientation of tool edge are shown, in Fig. 5(d). Plots in Fig. 5(d) are measured as shown in Fig. 5(b); angle λ , an angle between the rake face and a tangent to the contour line of worn tool; length δ , a distance from a point on contour line to a cross point with the initial tool line on perpendicular line to the contour line. Dot lines in Fig. 5(d) are drawn in the same way as shown in Fig. 5(c). In Fig. 5(c), the contour line of worn tool is arc of supposed circle. Radius of the supposed circle is determined to be equal the two extents of an oblique line (worn area) in Fig. 5(b) and (c). The deviation of contour line of the actual worn tool for supposed circular arc wear are possible to be compared in Fig. 5 (d).

These tool wear and wear shape are under the influence of the route

diagram of crystallographic orientations in Fig. 3. The one-to-one correspondence is impossible to be found between Fig. 3 and 5, because the wear or lapping characteristic have not been investigated perfectly in all area of diamond crystal. But, on the restricted area, they had been examined now separately in the previous reports^{1) 2)}; the representative three planes, $\langle 100 \rangle$, $\langle 110 \rangle$ and $\langle 111 \rangle$, and planes on outside line of unit spherical triangle. Therefore, on the partial lines that the route line of crystallographic orientation of tool edge agrees approximately with the outside line of unit spherical triangle, the relations between wear and orientation of tool edge are possible to be discussed qualitative. In order to simplify the planes and directions in discussions, let write equal to $\langle 100 \rangle$, $\langle 110 \rangle$ and $\langle 111 \rangle$ planes, $\langle 100 \rangle$ and $\langle 110 \rangle$ directions.

The orders of easiness to wear or lapping (crystal plane/direction) are indicated by the previous examinations as follows; $\langle 110 \rangle / \langle 100 \rangle$, $\langle 100 \rangle / \langle 100 \rangle$, $\langle 110 \rangle / \langle 110 \rangle$, $\langle 100 \rangle / \langle 110 \rangle$, $\langle 111 \rangle / [\bar{1}\bar{1}2]$ and $\langle 111 \rangle / [\bar{1}\bar{1}\bar{2}]$. The plane and direction of flank position on orientation routes of tool T412 and T422 in Fig. 3 are close to $\langle 110 \rangle / \langle 100 \rangle$ and $\langle 110 \rangle / \langle 110 \rangle$ respectively. Therefore, these flank wear occurred large. That of flank position on route of tool T411 is $\langle 100 \rangle / \langle 100 \rangle$, and flank wear is low comparatively. That of flank on tool T421 is $\langle 100 \rangle / \langle 110 \rangle$, and flank wear is lowest. In the face of tools, the same ways are possible to be discussed. That of face on the tool T411 and T422 are $\langle 100 \rangle / \langle 100 \rangle$ and $\langle 110 \rangle / \langle 110 \rangle$, and these face wears are large relatively. In tool T412, that of face is $\langle 100 \rangle / \langle 110 \rangle$, and no face wear almost occurs. That of face on tool T421 is $\langle 110 \rangle / \langle 100 \rangle$, and the face wear occurs slightly.

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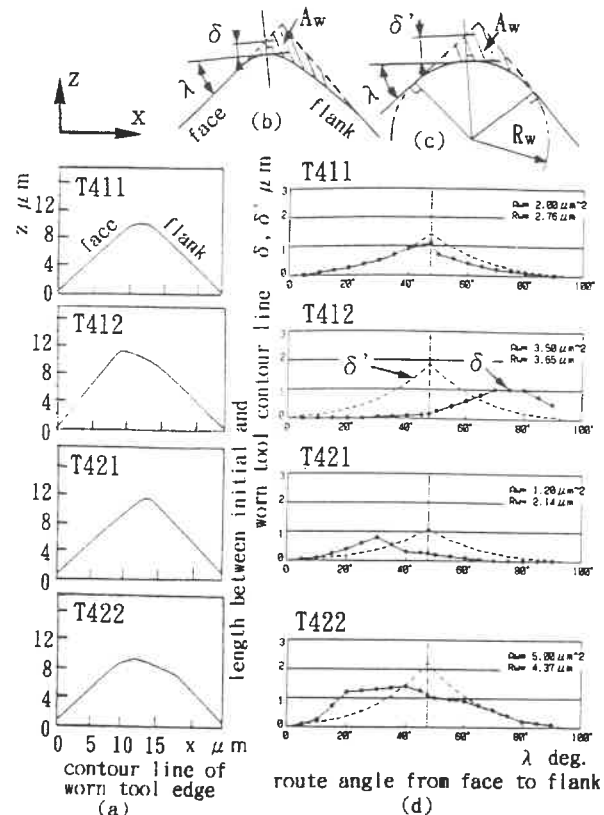


Fig. 5 Topographies of tool edge wear and relations to route angle from face to flank

Computer Controlled Precision Optical Polishing on the Diamond Turning Machine

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INTRODUCTION:

This abstract reports the force and wear data required to predict the material removal, or wear, for the Numerical Controlled (N/C) polishing program. The program's aim is to provide the operator of a N/C diamond turning machine or N/C grinding machine with the wear characteristics necessary to achieve uniform material removal.

The first phase of the program looks at a rotating polishing wheel, moving from near the center to edge of a rotating glass disc (fig.1). Future phases will look at more complex shapes.

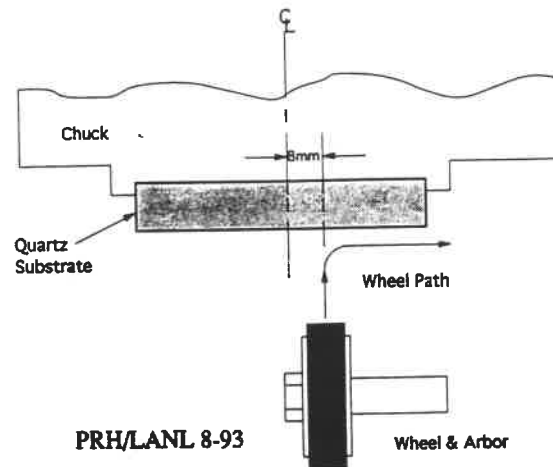


Figure 1 - Disc polishing detail

The amount of abrasive wear is proportional to the product of the pressure the polishing wheel exerts on the surface times the distance the surface travels relative to the polishing wheel [1]. This relationship expressed in equation form is known as Preston's equation:

$$\Delta H = (K)(P)(\Delta S) \quad (1)$$

Where,

ΔH = The amount of wear, m

K = Inverse of specific energy required to remove material, m^3/J

P = The pressure exerted by the polishing wheel, N/m^2

ΔS = Distance the sample travels on the polishing wheel, m

DETERMINATION OF K PARAMETER FOR OUR PROCESS:

In order to predict these wear characteristics, based simply on Preston's equation, the value of K for our process must be known. So an experiment was performed to determine this value.

We used a polyurethane polishing wheel and a 75 mm diameter quartz substrate or disc. The wheel had a partial toric contour with a major radius of 200 mm and a minor radius of 18 mm.

The precision N/C diamond turning machine was a Moore M-18 AG, located at the Target Fabrication Facility, Los Alamos National Laboratory. The machine was equipped with an air bearing polishing head (fig.2) which was in turn mounted on a Kistler 3-axis force dynamometer (fig. 3). Force data from the dynamometer was amplified by a matching Kistler amplifier and then stored in a Nicolet storage oscilloscope. Figure 2 gives a detail of the polishing head/quartz substrate (75 mm dia.x 12.5 mm, S/N 5) relationship, also shown is the position of the wheel plunges or polished zones on the substrate.

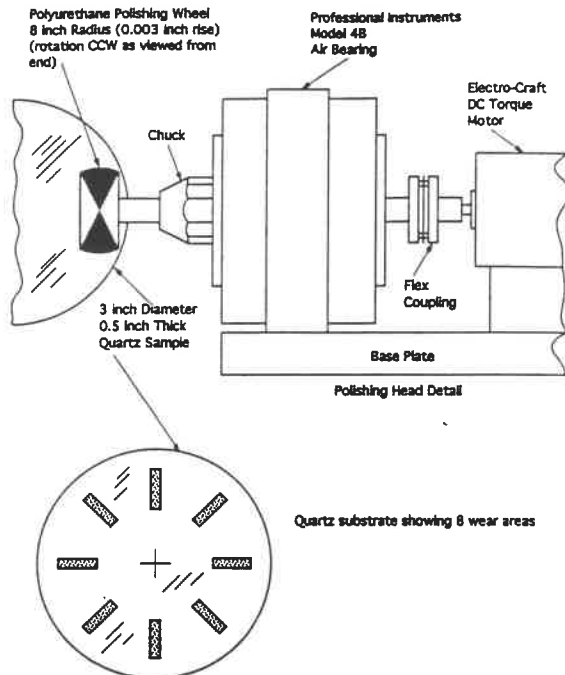
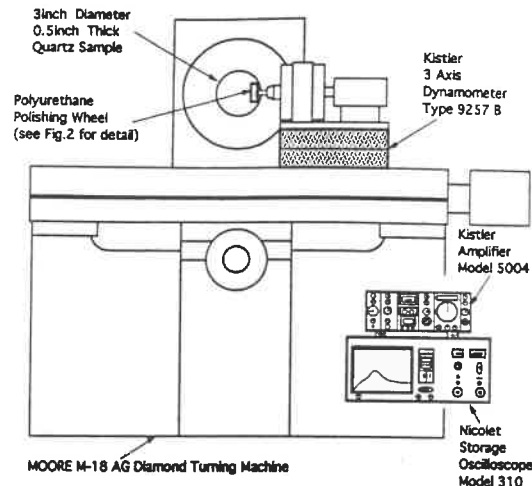


Figure 2 - Polishing head and wear test detail



PRH/LANL 8-93

Figure 3 - Experimental Set-up

These polished zones were created by the polishing wheel rotating at 1000 rpm being driven into the substrate to generate an average force which ranged from 1.2 to 6.7 N. The polishing wheel was continuously saturated with a cerium oxide polishing compound (Rhodite 906). The dwell times varied from 25 to 55 seconds.

A series of 8 polished zones were made. These zones (fig.2) were then measured with a Zeiss light sectioning microscope. Depth and area data provided information that enabled the prediction of the wear characteristics for this particular system. A plot of depth vs. pressure x distance is shown in figure 4. From equation 1 it can be seen that the slope of the line which best fits the data in figure 4 is the value of K for our process. The equation shown inside the plot in is the equation this line. The value for K is $1 \times 10^{-12} \text{ m}^3/\text{J}$.

PREDICTION OF SURFACE SHAPE BASED ON PRESTON'S EQUATION:

To perform polishing on a DTM an assessment of how the polishing process will affect the surface shape must be determined. To better understand this effect a model of the polishing process for our configuration was performed based upon Preston's equation. The process modeled, was that of: a single polishing pass across the surface of a disc with the wheel starting near the center (fig.1), and moving up to the parts edge.

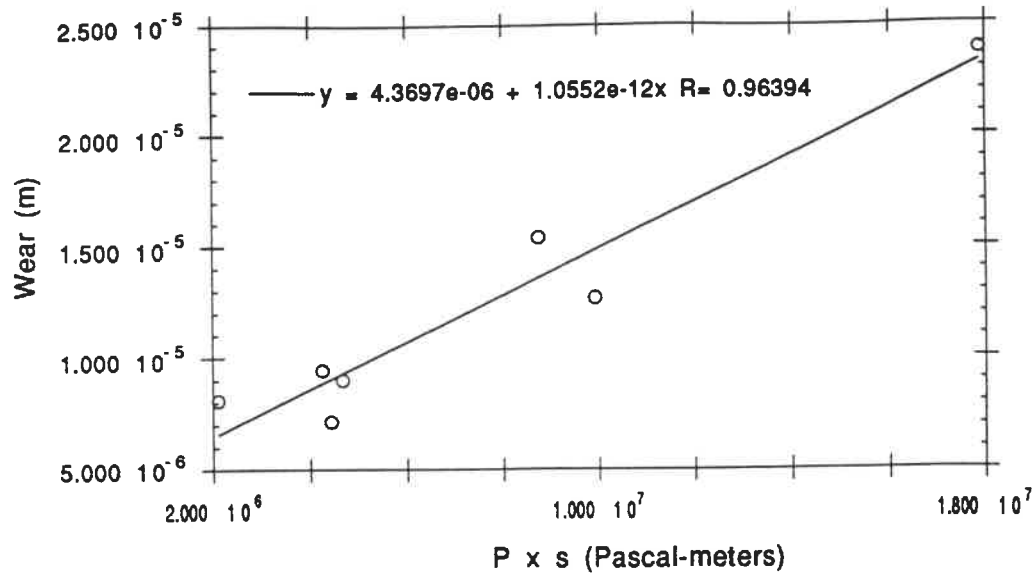


Figure 4 - Summary of wear data used to determine K for our process.

To aid in the modeling equation 1 must be rewritten in terms of velocity and time. The rate of wear is:

$$\Delta H / \Delta t = (K)(P)(\Delta S / \Delta t) \quad (2),$$

or since $\Delta S / \Delta t = V$, equation (2) can be rewritten:

$$\Delta H = (K)(P)(V)\Delta t \quad (3).$$

Where,

t = time, s

V = velocity, m/s

For our polishing configuration (see figure 1),

$$V = (r)(\omega_{wp}) + (R_{pw})(\omega_{pw}) \quad (4)$$

Where,

r = radius on the part where the polishing is occurring, m

ω_{wp} = angular velocity of the workpiece rad/s

R_{pw} = radius of polishing wheel, m

ω_{pw} = angular velocity of the polishing wheel, rad/s

and

$$\Delta t = \theta / \omega_{wp} \quad (5)$$

so

$$\Delta S = [(r)(\omega_{wp}) + (R_{pw})(\omega_{pw})](\theta / \omega_{wp}) \quad (6)$$

From figure 5 it is seen the approximation of $q = y / r$ may be made for small angles. (Even for angles as large as 30 deg this approximation may be used for our analysis without much error.)

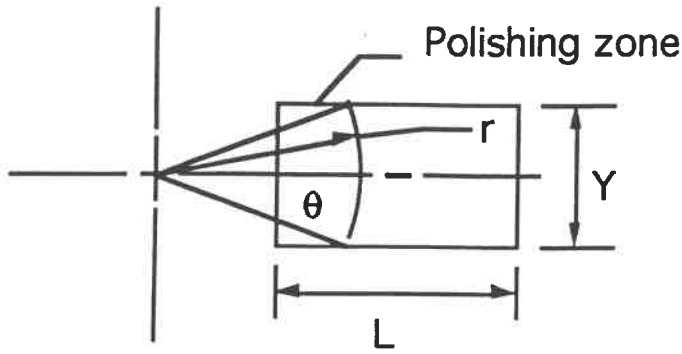


Figure 5 - Geometry of polishing zone

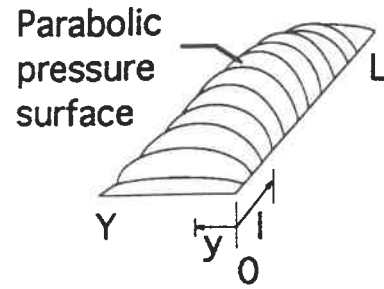


Figure 6 - Parabolic pressure profile

Substituting this relation into equation 6 and rearranging gives:

$$\Delta S = (Y)[1 + (R_{pw} / r)(\omega_{pw} / \omega_{wp})] \quad (7)$$

Now an expression for the pressure needs to be obtained. Measurements of the wear areas shown in figure 2 revealed that their 3-D profile looked much like the inverse of that shown in figure 6. A curve fit to the data indicated that depth was a parabolic function in both the y and l directions. This implies that the pressure is a parabolic function in these directions also. Since $P = 0$ at $l = 0$ and L , the expression for P becomes:

$$P(l) = (a)(l^2) - (a)(L)(l) \quad (8)$$

where,

a = a coefficient determined from the measured force, N/m^4

Since all radii must pass through the entire polishing zone in the y-direction, an average value for the pressure may be used for a given l-value. So the expression for $P(y)$ at a given l is:

$$P_l(y)_{avg} = \int P_l(y) dy / \int dy \quad (9)$$

For our situation of $P = 0$ at $y = 0$ and Y , and $P(Y/2) = P_{max} = P(l)$, then:

$$P_l(y)_{avg} = (2/3)P(l) \quad (10)$$

Combining equations 1, 7, 8, and 10 gives:

$$\Delta H = (K) (2/3)[(a)(l^2) - (a)(L)(l)] [(1 + (R_{pw} / r)(\omega_{pw} / \omega_{wp})](Y) \quad (11)$$

Equation 11, however, is the expression for only one pass through the polishing zone. The total wear is:

$$H = \sum \Delta H = (N)\Delta H \quad (12)$$

Where N is the number of passes a point on the workpiece makes through the polishing zone. The radial distance the polishing wheel will travel along the workpiece per revolution of the workpiece is:

$$\Delta l = f / v$$

where,

f = the feedrate of the polishing wheel, m/s

v = the frequency of a point moving into the polishing zone, Hz

Now

$$N = 1 / \Delta l = \sum \Delta l / \Delta l = (v / f) \sum \Delta l \quad (13)$$

Combining equations 11, 12, and 13 gives:

$$H(r) = (v / f)(K) (2/3)[(1 + (R_{pw} / r)(w_{pw} / w_{wp})](Y) \sum [(a)(l^2) - (a)(L)(l)] \Delta l \quad (14)$$

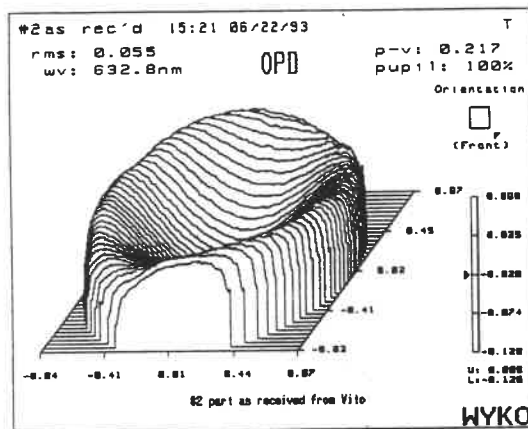
or as $\Delta l \rightarrow 0$

$$H(r) = (v / f)(K) (2/3)[(1 + (R_{pw} / r)(w_{pw} / w_{wp})](Y) \int [(a)(l^2) - (a)(L)(l)] dl \quad (15)$$

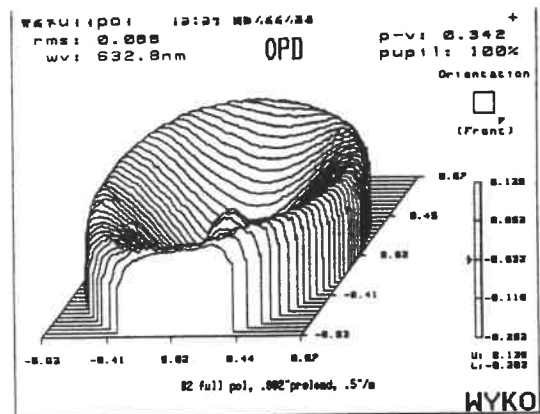
Equations 14 and 15 are the final ones used to describe our polishing process.

EXPERIMENT TO TEST MODEL:

The next test described here was designed to test the model of our process described above. Another 75 mm diameter disc of quartz (S/N 2) was placed on the diamond turning machine as before. Figure 7a shows a 3-D plot of the surface prior to polishing.



(a)



(b)

Figure 7 - Shape of disc #2's shape: (a) before polishing, (b) after polishing.

For the test, the Quartz disc was rotated at 100 rpm. The polishing wheel was rotated at 1000 rpm with the same compound saturation rate as before, the average polishing force was measured to be 3.6 N. The X-axis feed rate or wheel lateral movement was at 12.5 mm per min. A single pass was made with the wheel starting near the center (fig.4), and moving toward and past, the parts edge.

Upon completion of the test the part was evaluated. Figure 7b is the after polishing 3-D and cross section plot. The reader should keep in mind that the overall peak to valley error in these before and after plots do not exceed 0.35 waves (@6328Å) or 2.2 μ . Figure 9 shows the result of subtracting the before-polishing substrate surface from the after-polishing surface.

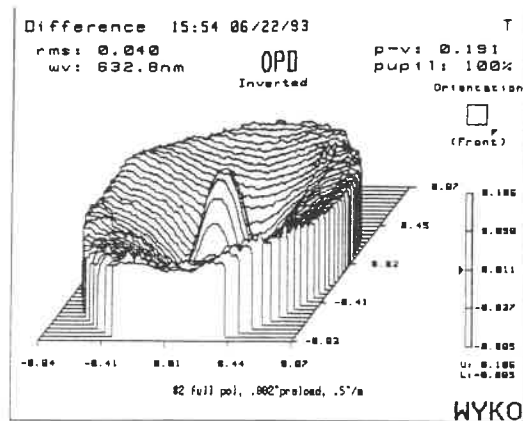


Figure 8 - Subtraction of disc #2's shape before and after polishing

A computer program was written to calculate the shape based upon equation 14. For this program: $a = -5.36 \times 10^9 \text{ N/m}^4$, $Y = 3.5 \times 10^{-3} \text{ m}$, $L = 12 \times 10^{-3} \text{ m}$. Figure 9 is a plot of the calculated and measured wear. The plot shows that the shape of workpiece after polishing can be predicted however the amount of material removal is not well predicted. This difference could be caused the way the value for K was measured, i.e. the workpiece was not turning when we made our plunges as shown in figure 2. Additional tests will be performed to resolve this discrepancy.

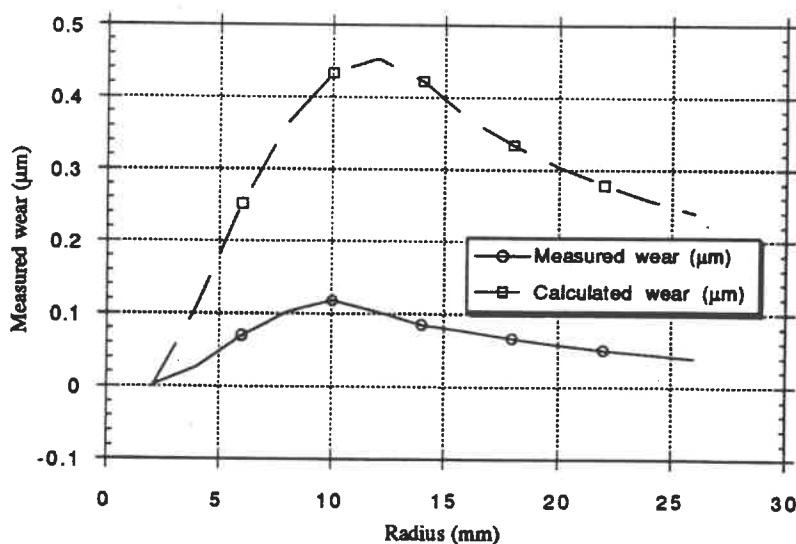


Figure 9 - Comparison of calculated and measured wear

CONCLUSIONS:

A model has been developed, using Preston's equation, to predict the shape of a workpiece after polishing on a N/C DTM. The model predicts the workpiece's general shape but does accurately predict the material removal. Additional work is needed to resolve this discrepancy.

ACKNOWLEDGMENTS:

The authors greatly appreciate the help of Dick Rhorer in setting up the Kistler Dynamometer.

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Figure Correction by Segmented Pressurizing Polishing Machine

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1. Introduction

For example, in a field of optical components or semiconductor wafers there is no bounds for demand of high performance. In some cases we will have to mass-produce those high quality articles which has both smooth surface and high form accuracy. In the mass production some degree of yield rate is also needed. One solution for dealing with that is advancing the final figuring process such as polishing. However it is not so effective nor economical in the case that a lot of high quality articles are needed. The other solution is applying combined process of usual (polishing) process and a special figure correction process. Only rejected articles after testing will be sent to the figure correction process and we can get high yield rate as a result. This way can be effective and economical in the case that severe specification is realized. In our study a new method of the figure correction process is discussed.

The figure correction process has been used in final figuring at manufacturing large optical component. The conventional process is controlling dwell time of a tool which is tiny in compare with the article. The tool is dwelled for a long time at zones which shows a big deviation from an expected figure. A matter of the method is taking a long time to be corrected because of the small tool size.

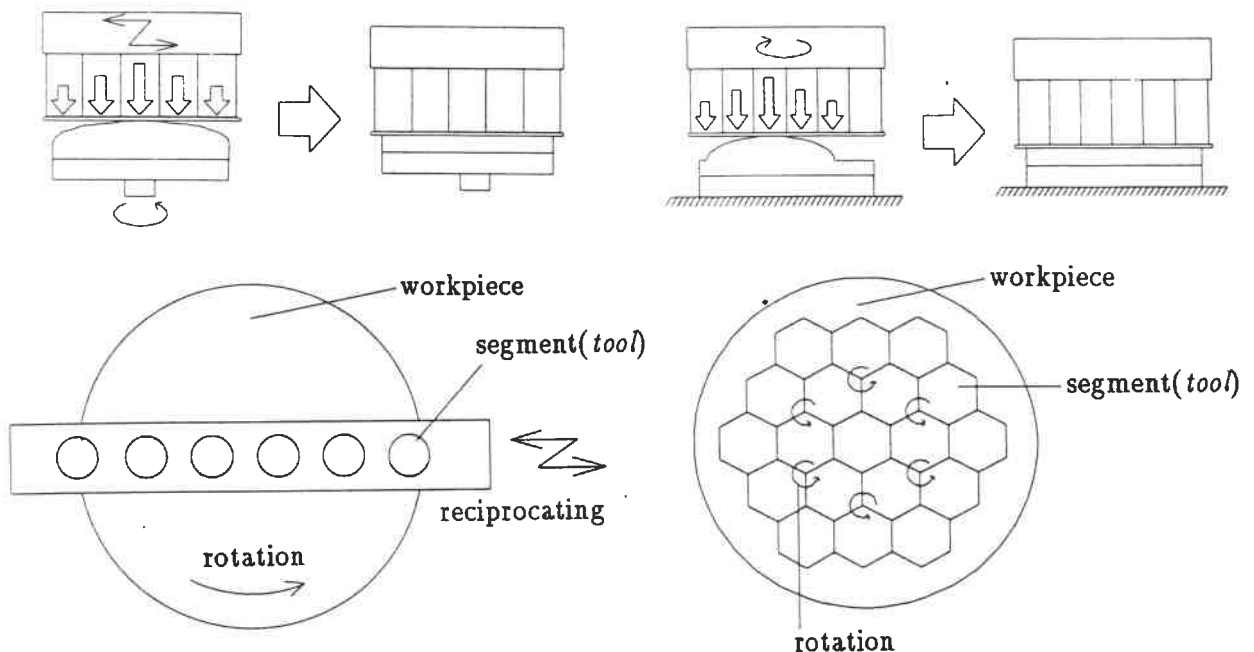


Fig.1a Linear arrangement of segmented tools

Fig.1b Array arrangement of segmented tools

Table 1 Comparison of tool arrangement

		linear type	array type
tool arrange dimension		1	2
number of segments		fewer	more
degree of freedom in motion		≥ 2	≥ 1
efficiency		less	good
controlability	(rotary symmetric)	easy	easy
	(rotary asymmetric)	complicated	easy
pressurizing		easy	difficult
abrasive supply		easy	difficult

In our method segmented tool in which we can apply pressure to each segment separately is used. The each segment is similar to the small tool in the conventional correction method and thus multiple effect is expected.

2. Consideration about segmented pressurizing machine

The following equation is the basic concept of the correction method which is same as the conventional, single tool's, one.

$$x \propto v * p * t \quad (0.1)$$

where x is material removal amount, v is relative velocity between workpiece and tool, p is applied pressure and t is machining time. In this chapter three problems about our multiple method, which are tool arrangement, way of pressure and figure controlling principle, are considered.

2.1. Tool arrangement

Two possible arrangements are compared. One is a linear type and the other is an array type. Example of each type is shown in Fig.1a and Fig.1b respectively.

In Table 1, comparison between them is shown. In the array type workpiece is almost covered by tools, thus slight motion is enough to execute polishing. On the contrary in the linear type as the tool does not cover the whole workpiece, both positioning and relative movement is needed. At Fig.1a those are workpiece rotation and tool reciprocating.

The array type tool has better efficiency and is easy to correct an asymmetric errors. It needs, however, so many segmented tools that pressurizing and supply of abrasive becomes rather difficult.

2.2. Way of pressurizing

Mechanical spring, electro-magnetic force and compressed air are applicable to be loaded the tool. The mechanical spring is hard to be controlled precisely with stability among plural tools. In electro-magnetic force, our calculation indicated that we have to apply rather big current when we intend to keep some amount of pressure so as to execute feasible machining speed. Therefore, problem caused by heat generation will be occurred. In that sense, the air is thought to be the best by its flexibility and controllability.

2.3. Figure controlling principle

From the equation (1), the fact that material removal can be controlled either by the

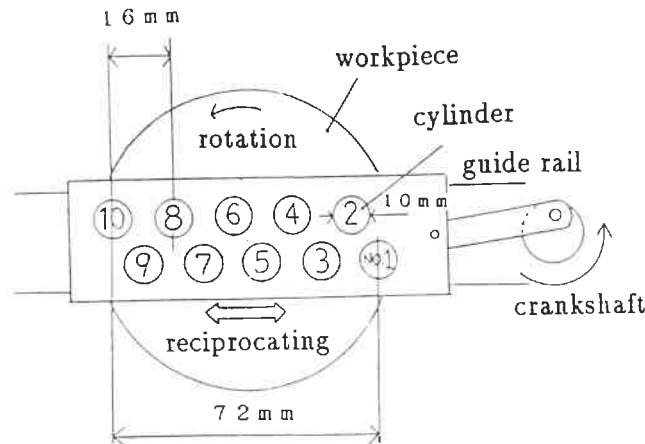


Fig.2 Tool arrangement of proto-type machine

applied pressure or by the time is found. Thus either pressure control principle in which each segment is pressured continuously by different amount, or on-time control principle in which each segment is pressured constant and is switched on or off individually, can be applied. With either principle, however, total correction time is almost same because the most deviated place in the workpiece must be machined continuously with possible maximum pressure in order to be removed most.

3. Experimental demonstration

3.1. Design principle of apparatus

We made a hand-made proto-type machine so as to prove this method. In designing the apparatus we selected a linear type segmented tool which is pressurized by compressed air through on-off type electro-magnetic valve controlled by personal computer. The figure correction is done by switching the each electro-magnetic valve with appropriate time sequence. The machine has 10 cylinders which are 5 pairs.(Fig.2) In each pair the tool is located symmetric. One of features of the method is that covering region of each individual tool is overlapped with that of their neighbor's tool. Beneath the cylinders polishing cloth is placed. The tool holder is reciprocated and the workpiece is rotated by motors respectively.

3.2. Software

Polishing strategy by this method is rather complicated. We invented an algorithm to get the time sequence of switching each segment. That is a kind of simulation program. We took into account of tool overlapping and elimination of excessive removal. In the software predicted shape after correction is also derived. As there is not any assurance that our algorithm is the best, study about that will be done more.

3.3. Machining results

In a experiment BK7 glass was used. The cross sectional shape of workpiece before the correction is seen in Fig.3. Flatness of it is more than $10\mu\text{m}$. By our software we derived the polishing sequence which can indicate times which each tool must be pressurized. Table 2 shows a result made by our program. In each pair, for example 1 and 10 in Fig.2, the time sequence is equal to one another. As the figure under the test was concave as is shown in Fig.3, outermost segment tools must be working during the most time and inner segment tools need less time. The predicted shape after the correction machined by the time sequence of Table 2 is also shown in Fig.3.

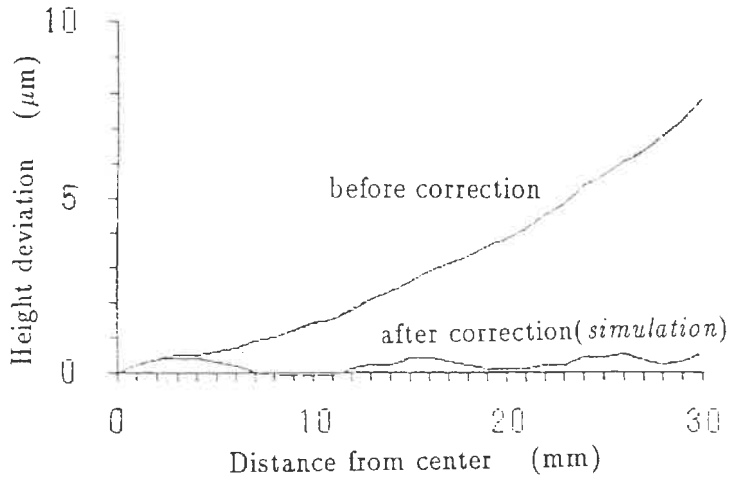


Fig.3 Simulation results

Table 2 Time Sequence

Tool num.	Time
1 & 10	475min
2 & 9	310min
3 & 8	205min
4 & 7	115min
5 & 6	0min

Table 3 Experimental conditions

workpiece material	BK7 glass
workpiece diameter	90mm
abrasive	cerium dioxide
polishing fluid	deionized water
polisher	artificial leather
polishing pressure	9.8kPa
workpiece rotation speed	55rpm
tool reciprocating frequency	200 rpm
tool reciprocating amplitude	12mm

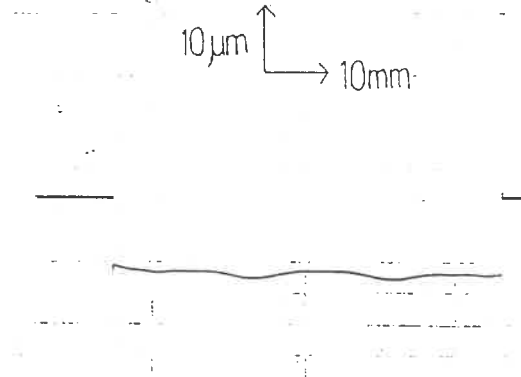


Fig.4 Machining results

We did an actual correction by that time sequence. Machining conditions are shown in Table 3. Machining results after the actual correction is shown in Fig.4. The actual corrected shape is almost coincide with the predicted one on amplitude of flatness which is less than $1\mu\text{m}$ and on number of fluctuation, which has almost three tiny peaks in the shape. Time needed in total was 7hours and 55minutes. An estimated time of the correction operated with a single tool was 36hours and 50minutes. Thus our method was 4.65 times more effective than the conventional one.

4. Conclusion

We think this correction technique can be successfully applied to generating an aspheric surface from a spheric surface. Moreover, if we add a rotary encoder to the workpiece turntable axis in order to control switching of electro-magnetic valve according to the angular displacement, we can correct an asymmetric figure error even by the linear type segmented tool.

SUPER FINISHING PROCESSES ASSISTED BY MAGNETIC FIELDS

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1. Introduction

With the rapidly increasing demands of precision machines with high speeds and improved performance, surface finishing becomes more crucial. To obtain super finished elements has been a vital important task for many manufacturing engineers. The super finishing processes assisted by magnetic fields are newly emerged technology to meet such manufacturing needs. In these processes, magnetic fields are introduced to enhance the machining capability as well as to control the machining processes. The advantages of introducing magnetic fields into these processes include small and uniform machining forces, a huge number of cutting edges participating in machining, and soft work-abrasive contacts, which consequently results in good surface integrity and dimensional accuracy of both metal and ceramic workpiece with complex geometries.

Most of the research and development on the subject so far has been concentrated in Japan [1] although the first invention was patented in the U.S. in as early as 1897 [2]. There have been hundreds of technical papers and reports published, and patents disclosed mostly in Japanese language since 1980, which are considered to be valuable assets to both manufacturing researchers and engineers. Fig.1 lists the technical invention and progress of the super finishing process in the U.S., Japan, and Europe in the chronological order. Fig.2 shows the various applications of different super finishing processes. In terms of the finishing media, these processes can be classified into three categories: magnetic abrasive finishing (MAF), magnetic fluid polishing (MFP), and magnetic float finishing (MFF).

2. Force and Motion Control Analysis

Cutting action occurs when abrasives move relative to workpiece surface. A magnetic field assisted polishing process uses

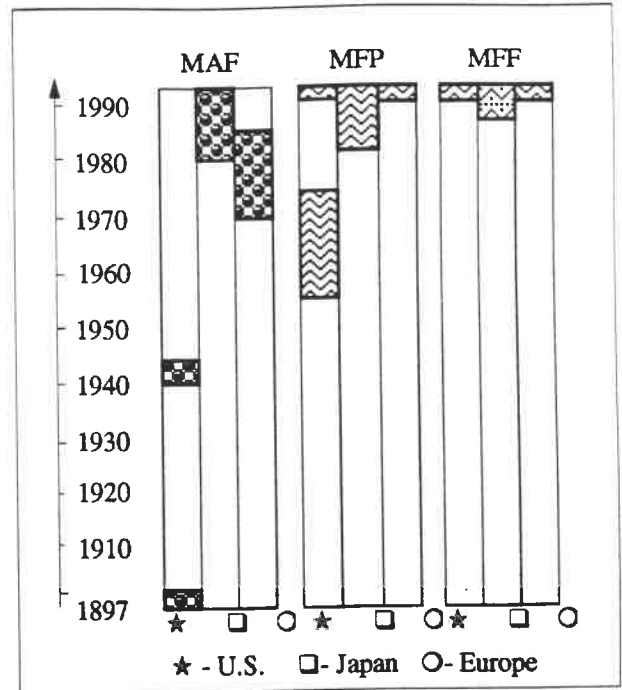


Fig.1 Technical invention and progress

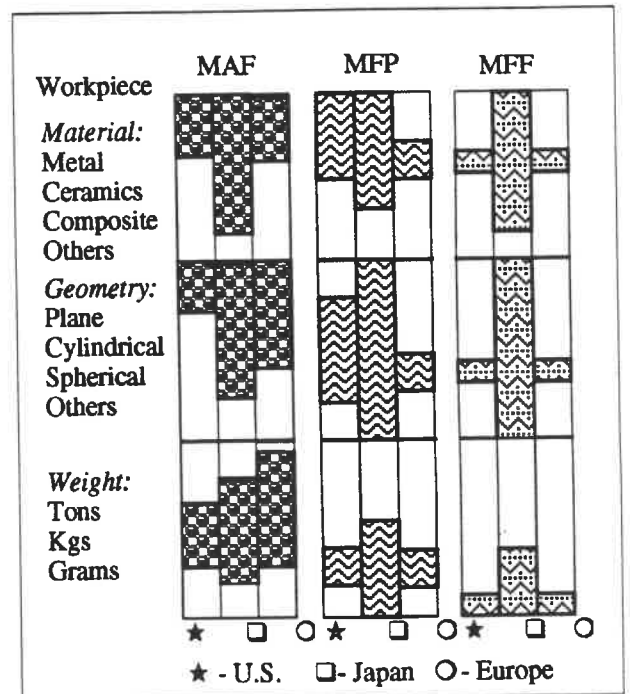


Fig.2 Applications of MAF, MFP, and MFF

magnetic field to force magnetic media to discharge towards workpiece surface to achieve material-removal. In contrast, in a grinding process, rigid bonding materials are used to force abrasives to penetrate into workpiece surface. There are always some abrasives that project too much from the wheel surface. The gentle and uniform contacts between abrasives and workpiece surface can be controlled by magnetic field to achieve good surface integrity. The workpiece-abrasive contacting forces are proportional to magnetic field strength and its gradient that can be varied by working clearance and other dimensional parameters of workpiece and magnetic poles. The magnetic force F_z acting on workpiece surface towards lower side of magnetic field is given as,

$$F_z = \mu_0 V_m (\chi_w - \chi_m) \left[H_x \frac{\partial H_x}{\partial z} + H_y \frac{\partial H_y}{\partial z} + H_z \frac{\partial H_z}{\partial z} \right]$$

where μ_0 is magnetic constant, V_m the volume of magnetic abrasives, H strength of magnetic field, χ_w and χ_m are susceptibilities of workpiece and magnetic medium.

In polishing ceramic balls, in order to get good surface integrity and high polishing rate, polishing forces on individual balls, and ball movement are to be effectively controlled. The contact between balls and polishing plates are assumed to be elastic, with most of deformations on plates. The basic motions of the ball relative to the polishing plate include rolling and spinning as shown in Fig.3. The sliding motion may also occur which is harmful to the formation of ball roundness and sphericity. If the ball only rolls forward without spinning and sliding, less cutting will be achieved near the minor axis of the contact zone, resulting in low material-removal rate. If the ball only spins, high material-removal rate can be obtained. However, larger and uneven wear of polishing plate will also be accompanied. Proper combination and control of rolling and spinning motions are a must to achieve high polishing rate, good geometric accuracy and surface integrity. The speed ratio of ball rolling and spinning motions should be determined in accordance with the material properties of workpiece and polishing plate. The motion control of the ball in Fig.4

can be accomplished as follows:

$$\theta = \tan^{-1} \frac{(R_C \omega_C - R_A \omega_A) \sin \alpha - (R_B \omega_B - R_A \omega_A) \sin \beta - (R_B \omega_B - R_C \omega_C)}{(R_C \omega_C - R_A \omega_A) \cos \alpha + (R_B \omega_B - R_A \omega_A) \cos \beta}$$

$$V_0 = \frac{R_B \omega_B \sin \beta - R_C \omega_C \sin \alpha - \tan \theta (R_B \omega_B \cos \beta - R_C \omega_C \cos \alpha)}{\sin \beta - \sin \alpha - (\cos \beta - \cos \alpha) \tan \theta}$$

$$\omega_b = \frac{R_B \omega_B - V_0}{r \sin(\alpha + \theta)}$$

where ω_b is the rotation speed of the ball, V_0 the translation speed of the ball center and θ an angle determined by the dimensional and speed parameters of the plates. The rolling speeds Ω_{Ar} , Ω_{Br} , Ω_{Cr} , spinning speeds Ω_{As} , Ω_{Bs} , Ω_{Cs} , and ball center translation speeds V_{OA} , V_{OB} , V_{OC} , relative to the plates A, B and C are the following:

$$\Omega_{Ar} = \omega_b \cos \theta$$

$$\Omega_{Br} = \omega_b \sin(\theta + \alpha) - \omega_b \cos \alpha$$

$$\Omega_{Cr} = \omega_b \sin(\beta - \theta) + \omega_b \cos \beta$$

$$\Omega_{As} = \omega_b \sin \theta - \omega_A$$

$$\Omega_{Bs} = \omega_b \cos(\theta + \alpha) - \omega_b \sin \alpha$$

$$\Omega_{Cs} = \omega_b \cos(\beta - \theta) - \omega_C \cos \beta$$

$$V_{OA} = V_0 - R_A \omega_A$$

$$V_{OB} = V_0 - R_B \omega_B$$

$$V_{OC} = V_0 - R_C \omega_C$$

Therefore, by controlling the speed ratios of the polishing plates, and the strength of magnetic field, ball workpiece can be polished with improved roundness, sphericity, surface integrity, and at increased polishing rate.

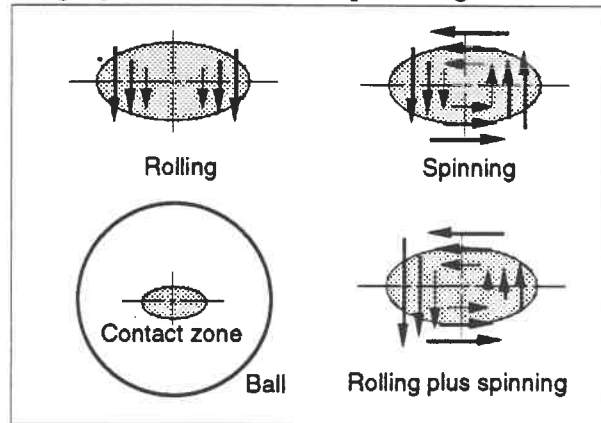


Fig.3 Various speeds and motions in ball polishing

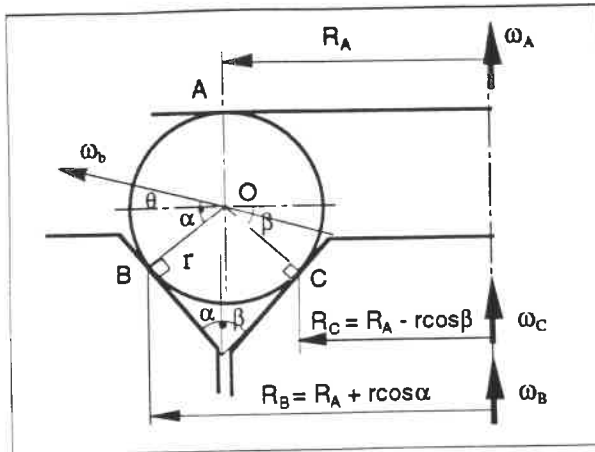


Fig.4 Three contact zones in MAF

3. Magnetic Abrasive Finishing (MAF)

Magnetic abrasive agglomerates are filled into the gaps between workpiece and the magnetic poles as shown in Fig.5. The agglomerates are 50 - 100 μm in size that can be obtained by sintering ferrous material powders with 1- 10 μm abrasives [3]. The polishing pressure is exerted on workpiece surface through the agglomerates by the applied magnetic field. The agglomerates are linked to each other magnetically along the lines of magnetic force, forming flexible magnetic abrasive brushes.

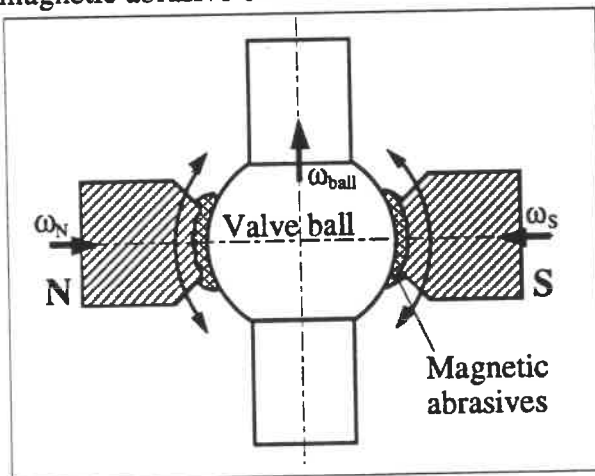


Fig.5 Magnetic abrasive finishing

In a single-point diamond turning process, the tool-workpiece contact zone is a point, and in a finish grinding process, the contact zone is nearly a line, while in a magnetic abrasive finishing process, the contact zone is a surface. Much more cutting edges are involved in MAF

process than those in any other processes. A grinding process has poor controllability over cutting forces of individual abrasives. This often causes unexpected scratches on workpiece surface. As compared to the rigid bonding of a grinding wheel, the flexible bonding in MAF process provides each individual abrasive with uniformly controlled cutting forces, which will result in good surface integrity of workpiece. This unique feature allows the magnetic abrasive finishing process to undertake finish polishing effectively. For a cylindrical workpiece, surface and edge finishing can be accomplished in a time period much less than that by a conventional finishing process. Bearing steel rollers with a hardness of Rc 63 can be mirror finished in 30 seconds.

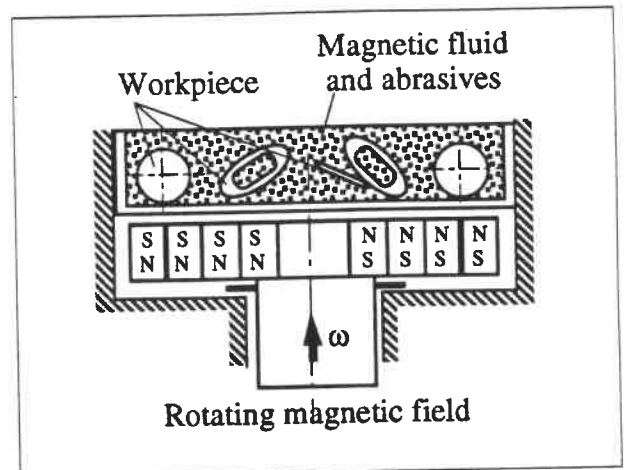


Fig.6 Magnetic fluid polishing

4. Magnetic Fluid Polishing (MFP)

Magnetic fluid is used in this polishing process to agitate abrasive slurries to polish workpiece. The fluid is retained in a fixed container in an alternating magnetic field as shown in Fig.6. Fig.7 shows a polishing application example of utilizing magnetic fluid to levitate non-magnetic workpiece immersed in magnetic fluid through an external magnetic field. Since the polishing pressure is obtained by a buoyant levitational force, the forces applied by the abrasives to the workpiece are small and highly controllable. The workpiece immersed in the magnetic fluid can be complex in geometry and grams in weight. For the case of polishing ceramic balls, an upper plate and a float pad are used to control

the balls to get higher polishing rate and more uniform surface finish. Time required to polish silicon nitride balls is only one to two hours which is one to two orders of magnitude less than that required by a conventional polishing process [4]. A typical result obtained on silicon nitride balls was $0.2\ \mu\text{m}$ in sphericity and $0.05\ \mu\text{m}$ in surface roughness.

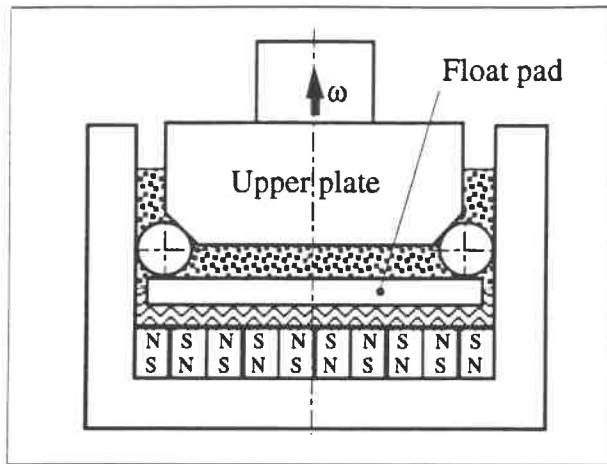


Fig.7 Magnetic fluid polishing

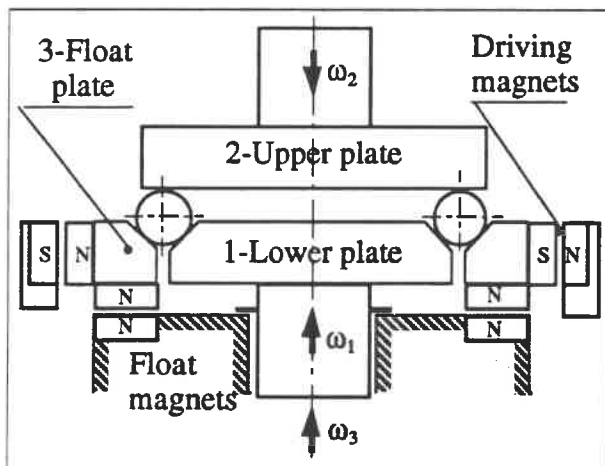


Fig.8 Magnetic float finishing

5. Magnetic Float Finishing (MFF)

The process utilizes magnetic repulsion to float workpiece against a polishing pad or a float pad against workpiece. The float pad automatically adjusts the distribution of polishing pressure to control the surface roughness, the geometry of workpiece, and the dispersion of workpiece. In finishing ceramic balls shown in Fig.8, magnetic repulsion

provides soft ball-plate contacts without damaging the balls even at high speed. The spherical workpiece can be precisely controlled by split V-groove plates. Because of the damping effect of the floating V-groove plate, the polishing process can achieve good workpiece surface integrity and dimensional accuracy. Typical results of polishing 24 silicon nitride balls by this process are as follows [5]:

	Before polishing	After polishing
Dispersion	$73\ \mu\text{m}$	$0.25\ \mu\text{m}$
Sphericity	$3\ \mu\text{m}$	$0.13\ \mu\text{m}$
Ra	$0.213\ \mu\text{m}$	$0.02\ \mu\text{m}$

6. Summary

The supper finishing processes assisted by magnetic fields are very especially efficient for finishing hard and brittle materials such as ceramics. MAF process allows a working clearance to be filled up with magnetic abrasives. The demands on the motion accuracy of the magnetic abrasive finishing machines and capital costs are low and the process can be easily automated in mass production. MFP process is a thermal stress free process. Workpiece with complex geometries can be effectively polished without mechanical distortion. The process can simultaneously polish workpiece of different sizes and shapes at a high polishing rate. MFF process has the ability of self-adjusting the polishing pressure to improve the surface finish and form accuracy of workpiece. Good surface integrity is achievable because any damages to workpiece can be avoided by the floating force.

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COMPUTER ANALYSIS OF THE KINEMATIC SYSTEM OF LAPPING MACHINES OF PLANES

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INTRODUCTION

The typical kinematic system of one-side lapping of flat surfaces is an annular system[6]. In this solution of the executory system, the conditioning rings, amounted normally from one to four, perform the function of elements, equalizing the lapping disk and keep in a definite position the separators of the lapped object and are taking the required relative motion to the lapped objects. The wear uniformity of conditioning rings decides, among other things about the flatness condition of the lapping disk, and thereby affects the flatness of the lapped surfaces. Owing to the time stable velocity of the lapping disk, the motion of the conditioning rings is also stable. This is influenced by the conditions of interaction, occurring in the system: lapping disk - abrasive suspension - workpiece (or conditioning ring), so the suspension characteristic (grade, number and concentration of abrasive micro-grains; kind, dose and properties of the carrier), superficial pressure per unit area (dimensions and weight of workpieces and rings) as well as the material of the disk and conditioning rings. Generally, about the rings position in relation to the lapping disk centre, decides the machine operator.

In the result of simultaneous or individual radial displacement of the ring centres of wear in concern to the lapping disk. So, it is obvious to state that the lapping disk wear affects, first of all, the filling out manner of the separator by the lapped pieces. This state is conditioned as well by the arrangement of pieces in relation to the separator centre, their basic concentration, as by the profile of lapped surfaces. These factors have influence on the summary contact path of each point, laying on the active surface of lapping disk, so they decide about the radial profile (wear) of the tool.

While designing a separator, a decision must be taken, concerning the lay-out manner of pieces and on the other hand, owing to the kinematics complexity of the executory system of lapping machine [1, 2, 3, 4, 5, 7, 8, 9], the computer assisted designing becomes indispensable.

CHARACTERISTIC AND SOME SIMULATION RESULTS

The elaborated computer program provides certain shape freedom of lapped pieces and of their position on concentric circles in the separator. The greatest amount of pieces on one circle can not be greater than 50 off, while the amount of circles, on which the pieces are arranged, can not exceed 5. This range includes the possible practical situations to the produced lapping machines by the outstanding firms.

Data for the program include:

rdw - internal radius of lapping disk, mm
rdz - external radius of lapping disk, mm
rd - radius of lapping disk, mm
ro - separator centre distance from lapping disk centre, mm

rpw - internal radius of conditioning ring, mm
 rpz - external radius of conditioning ring, mm
 nd - rotational speed of lapping disk, r.p.m.
 np - rotational speed of conditioning ring (normally in practice $np=ns$), r.p.m.
 tdoc - time of lapping (of computer simulation), s
 rsj - j-radius of pieces laying-out in separator, mm
 pfji - angle defining the position of i-piece on j-separator circle, radian
 lej - number of pieces arranged on j-separator centre,
 w(k) - coordinates of piece vertexes with polygonal profile (in coordinate system, related to separator), mm.

The simulation program allows to analyse the speed v in relative system: lapped piece - lapping disk (or conditioning ring - lapping disk) and it determines the contact path s of pieces (conditioning rings) with disk points between its minimum (internal) and maximum (external) radius. The computing procedure allows also to analyse the collision probability in arrangement of the pieces.

The inserted, hereafter, by way of example, simulation results concern the single-disk lapping machine Lapmaster 36. On the figure 1 there are presented diagrams of relative velocity variation and contact path of the gu-

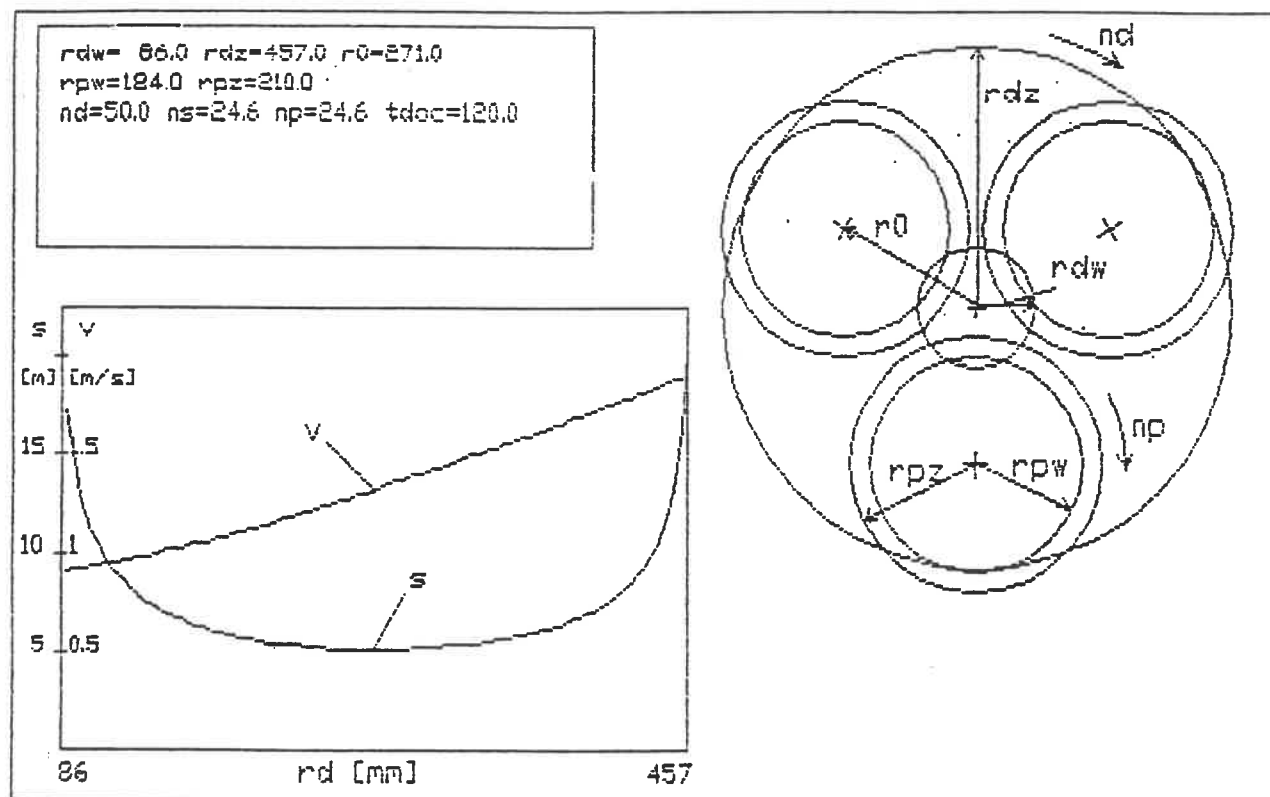


Fig. 1. Simulation results for conditioning ring of the lapping machine Lapmaster 36

iding ring in the executory system of lapping machine. Owing to the fact that the active width of rings must be relatively small, the greatest influence of ring motion on the disk wear characteristic, takes place at its external and internal edge, hence, a convex shape is formed.

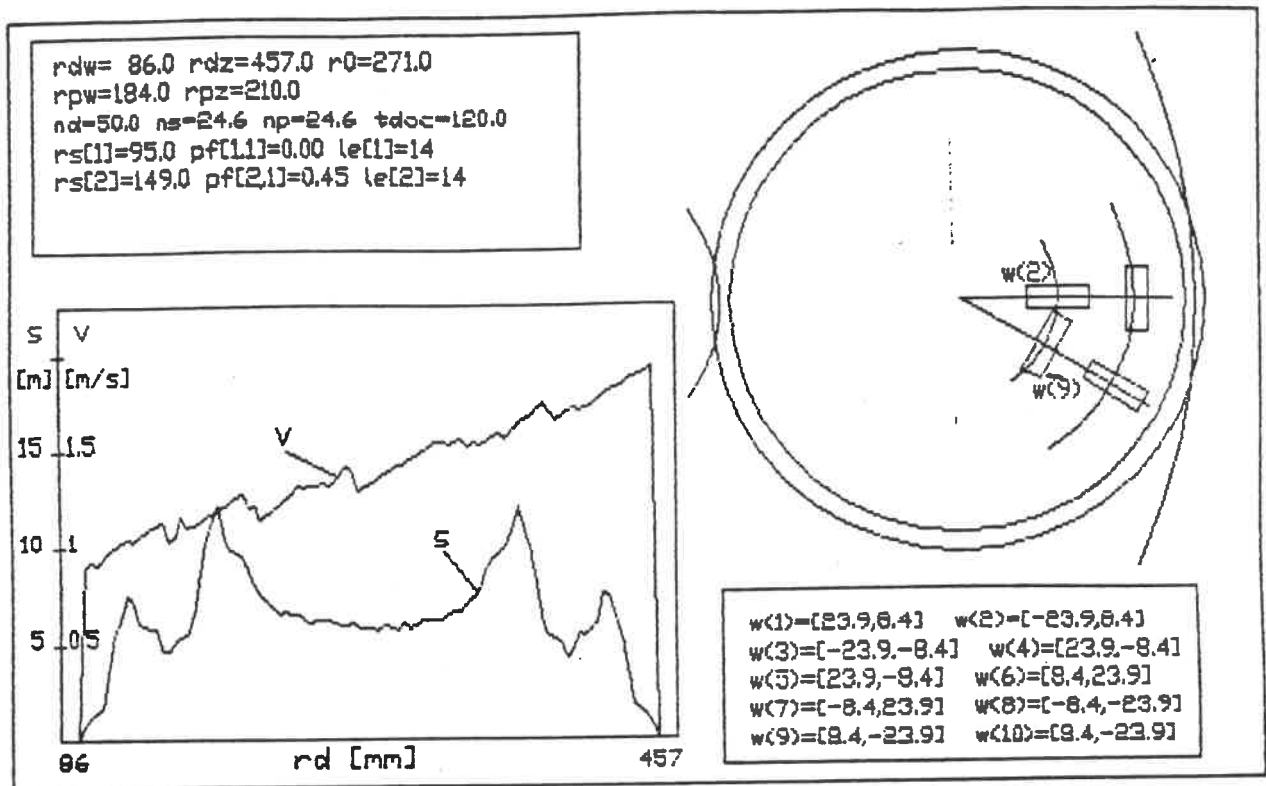


Fig.2. Simulation results for rectangular lapped workpieces

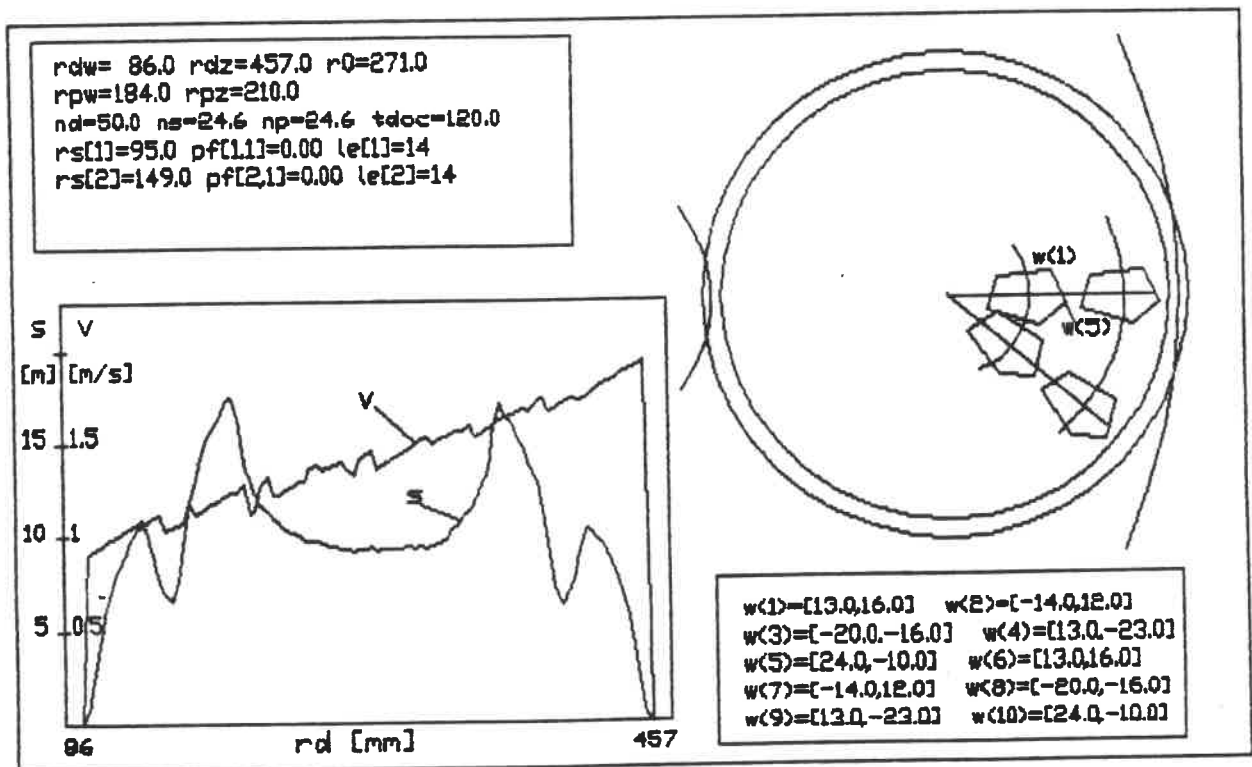


Fig.3. Simulation results for pentagonal lapped workpieces

In the analysis of polygonal piece arrangement, has been assumed a ratio stability of the circumference square and lapped surface area (corresponding to an equilateral triangle, and so $12\sqrt{3}$). It was also assumed stability of the lapped surfaces of each workpiece (800 mm^2) and the part stability of conditioning rings and filling with pieces (12%). It is obvious that the separator filling degree has influence on the time of calculations. The pieces with maximum pentagonal profiles were included in the simulation. Other actual forms must be approximated with pentagon. On figures 2 and 3 there are examples of calculations, for several profiles. The data concerning the program include a declaration of five vertexes of profiles, and so, in case of triangle, or quadrangle description, a principle of the last notations repeatability, must be observed. This situation is illustrated by means of included here examples.

It results from the carried-out analyses, that the path values of the contact of elements with points on the active surface of lapping tool in cross-section have a great range. In the central zone there is a local minimum. The level of s value depends on the small degree of the profile of the profile of lapped surface.

The greatest value of contact path was obtained on the smaller radius of the separator. The average speed of the lapping tool (for concerned machine), changes about 1.8 times and has a dependence form, close to linear one. In conclusion, a hypothesis may be assumed, that to decrease the contact path unevenness of the particular disk fragments with objects, it is necessary to change the drive method of rings (separators), or to introduce a variation of their rotation during one running cycle.

GENERAL REMARKS

The developed computer program allows in a large value range the analysis of geometrical, kinetic parameters of the single-disk lapping machine executory system and also workpiece profiles and their arrangement on the separators, as well as the separators themselves, on the lapping tool surface. It may be used for the optimization process of lapping machine executory systems and also while constructing the separators with circular and polygonal seatings. Having regard to the influence of lapping speed on the disk wear and quality of the treatment process, it is advised to continue the undertaken problems.

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Comparison of Measured and Predicted Plough Geometries and Forces for a Diamond Pyramidal Indentor Ploughing a Gold Surface

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INTRODUCTION

An understanding of the mechanical behavior of materials at nanometer scale dimensions can be of great technological importance. This knowledge may lead to the production of ultra precise engineered surfaces with applications which range from optics to catalysis to high speed electronics. The synergism of theory and experiment is the most effective way to gain this understanding. This paper describes the results of a comparison between the predictions of a rigid/perfectly plastic model of ploughing with experimental data for ploughing depths of about 20 nm.

DESCRIPTION OF PLASTICITY MODEL

The plasticity model uses the upper bound method for a rigid/perfectly plastic material [1]. This method is used to calculate the plough geometry and the ploughing forces by simultaneously enforcing zero material flux across a specified boundary and minimizing the rate of work needed to create the plough. The above requirements mean that there is no material flux across the planes ABC and ACDE as depicted in figure 1 (a). The model calculates the geometry of the displaced material, as shown in figure 1 (b), which satisfies the constraints.

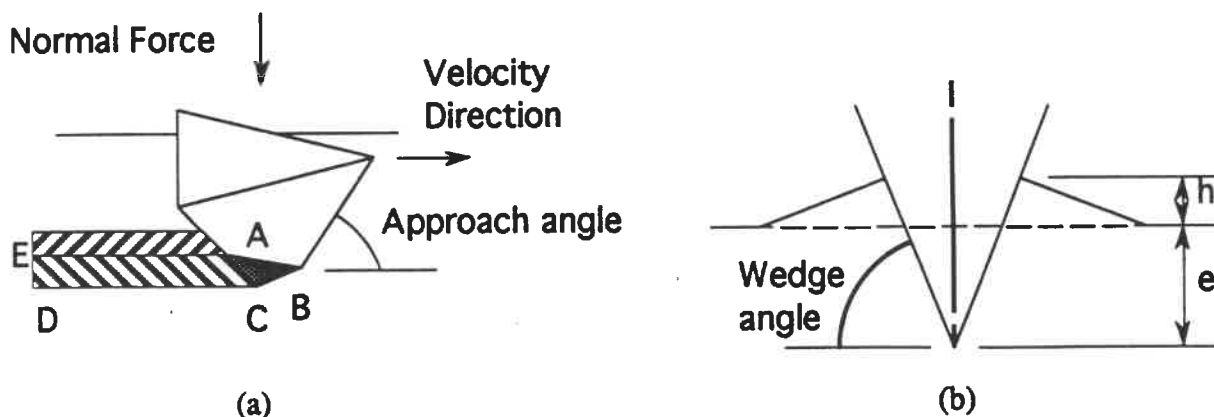


Figure 1 - Drawing of plough geometry used in the plasticity model.

The information needed to relate the model to the experiment is:

- 1) the indenter's geometry,
- 2) the normal ploughing force,
- 3) the plough depth and height, and
- 5) the yield strength of the material.

DESCRIPTION OF EXPERIMENT

A stand-alone Scanning Tunneling Microscope (STM) head was used to both produce the plough and measure its geometry. The ploughing sample was placed on the pan of a Mettler balance and the STM head was mounted on a fixture which rested on the frame around the pan. Figure 2 depicts this arrangement which allowed the measurement of the normal force exerted on sample by the STM's tunneling tip to be made to a resolution of 10^{-7} N.

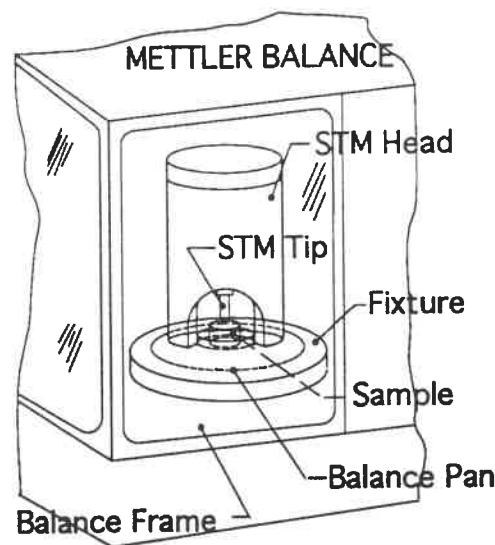


Figure 2 - Schematic of experimental configuration.

It was discovered during the process of measuring the tunneling force as a function of voltage and current [2], that enough force was induced by a doped single crystal diamond STM tip to plastically deform gold. It was then decided to use this tip, which was lapped into the shape of a three sided pyramid, as the indenter in a series of ploughing experiments.

The ploughing plane consisted of the (111) plane of a single crystal gold sample which had been lapped to a surface finish of 5 nm rms. (The surface finish was measured using a TOPO-3D® [3] optical profilometer with a 40x objective lens.)

As was mentioned above, the same tip used to plough the surface was also used measure the plough geometry for comparison with the calculated geometry. This, of course, can present metrological problems. The main geometrical comparison made is between the measured and calculated height to depth ratio (h/e as shown in figure 1b). Stilwell and Tabor [4] demonstrated that when a hard conical indenter is pressed into a metal (i.e. an elasto-plastic material) there is little change in the diameter of the indentation during withdrawal but there is a decrease in its depth. This means that the included angle of the indentation is more obtuse than the included angle of the indenter. So the indenter should be able to measure both the height of the displaced material and

the depth of the plough. However, if elastic recovery causes the depth of the indentation to be less than the depth actually created by the indenter, this can lead to an error. This error should be very small since hardness data taken with a Nanoindenter® [4] on the gold sample used for this set of experiments demonstrates that it has very little elastic recovery. This small amount of elastic recovery also means that the sample should fit the model's assumption of a rigid/perfectly plastic material.

To create a plough, the Y-direction of the imaging scan was disabled and the scan frequency was changed to 0.13 Hz. The tunneling voltage was then reduced from the normal imaging voltage of 1v to 1.4 mv at the beginning of a X-direction scan. When the tip had finished a single traverse the tunneling voltage was again increased to 1v. The force was recorded via a RS232 interface during the above process. A line plot of a 1 μm long plough is shown in figure 3. (The vertical axis amplification was increased by a factor of 6 to allow the plough to be viewed easily in figure 3. Consequently, the areas which appear very flat are where the data exceeded the values allowed for the plot.) Figure 4 shows a sections taken perpendicular to the ploughing direction. The section data was used to infer the indenter's geometry for a number of reasons: 1) the almost perfectly plastic nature of gold should leave a very good impression of the indenter's shape, and 2) it is difficult to measure the indenter's shape a few ten's of nanometers from its apex even with a Scanning Electron Microscope.

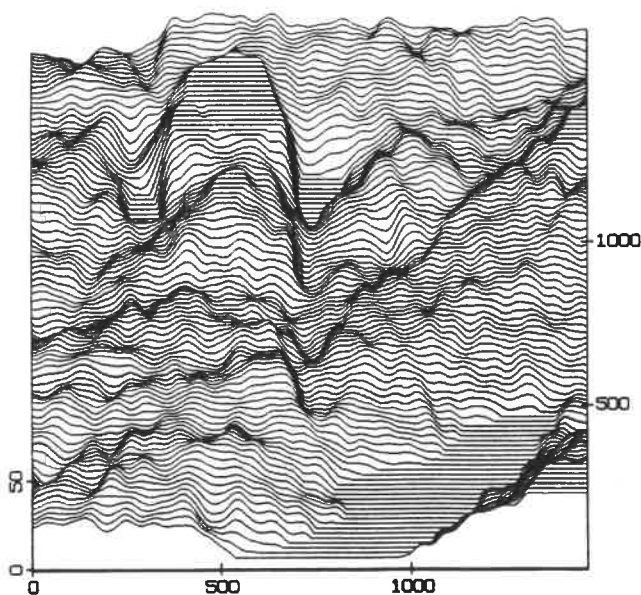


Figure 3 - 3-D line plot of a 1 μm long plough

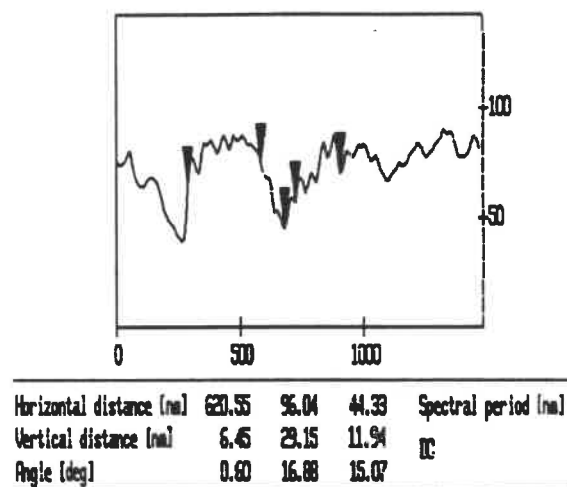


Figure 4 - Section taken through plough

An examination of figure 3 reveals that the displaced material is wavy in nature along the plough as opposed to being uniform in height. This causes some problems with the data analysis and is not predicted by the model. This wavy pattern could be caused by a combination of a number of effects which include: 1) the interaction of the slip systems in the material, 2) the variation in the friction factor along the plough caused by the interaction of the surface contamination layer with the indenter and gold surface, 3) mechanical instabilities in the apparatus, and 4) the fact that the sample surface was not perfectly flat initially. Because of this waviness an averaging approach was taken to the data analysis.

The approach angle was measured to be 20 degrees from a section taken through the plough valley. The average wedge angles were determined to be 24 and 16 degrees from two section views. A calculation of the normalized force and h/e (see figure 1b) for various friction factors is plotted for these two angles in figure 5. (A friction factor of 0 means there is no adhesion of the gold to the diamond. A friction factor of 1 means that there is perfect adhesion of the gold to the diamond.)

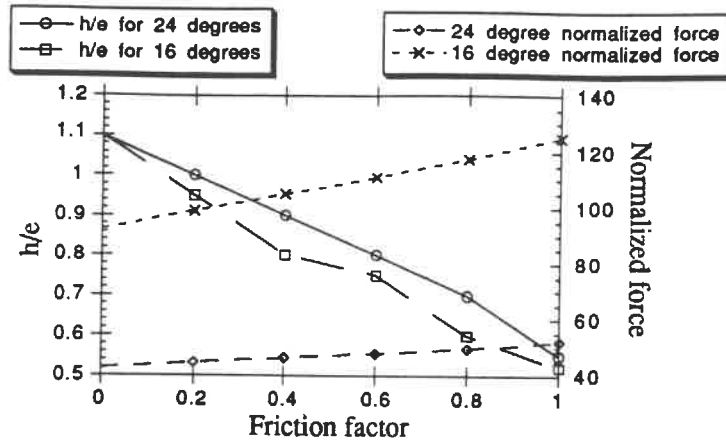


Figure 5 - Plot of the calculated height-to-depth ratio and the normalized force needed to produce a plough for wedge angles of 16 and 24 degrees at an approach angle of 20 degrees.

The vertical force is related to the normalized force and the yield stress of the material by equation (1):

$$\text{Vertical force} = (\text{Normalized force})(\text{Yield Stress})(e^2) \quad (1)$$

It is accepted that the yield stress of a material is approximately 1/3 of the hardness value [6]. A plot of hardness vs. depth for our sample is shown in figure 6.

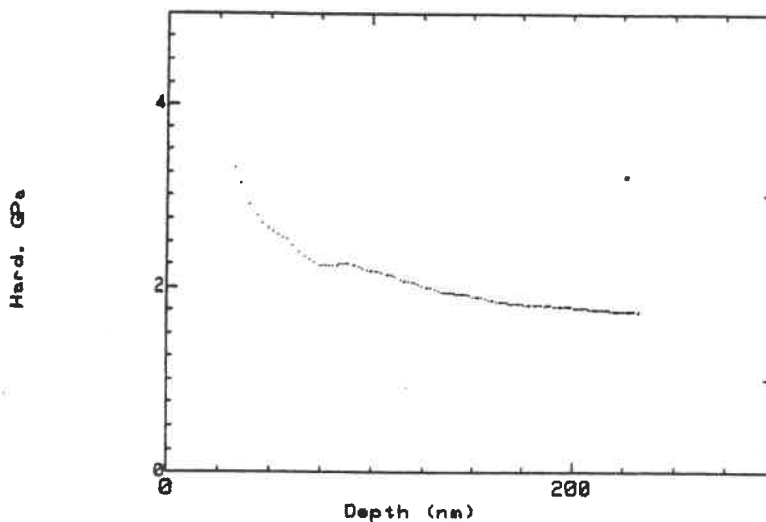


Figure 6 - Plot of hardness vs. depth for the gold sample.

Our measured depth (i.e. the value for e) is about 17 nm. Since no data is given at this depth, 3 GPa is used as a representative value. The yield strength of the gold is then estimated to be 1 GPa.

The average h/e value measured for the 24 degree angle was 0.52. From figure 5, the closest friction factor which corresponds to this has a value of 1.0. The normalized force at this friction factor is 52. For the 16 degree angle h/e was 0.85 which corresponds to a friction factor of 0.4 and a normalized force of 105. Using the above values in equation (1), gives

for the 16 degree wedge angle:

$$\text{Vertical force} = (105)(1 \times 10^9)(17 \times 10^{-9})^2 = 30 \times 10^{-6} \text{ N}$$

and for the 24 degree wedge angle:

$$\text{Vertical force} = (52)(1 \times 10^9)(17 \times 10^{-9})^2 = 15 \times 10^{-6} \text{ N}.$$

The measured force was 8.1×10^{-6} N which is a factor of 2 to 4 times less than the calculated value. It should be noted, however, that the absolute values for hardness measured at depths less than 50 nm are subject to question [7] and a hardness of 2 GPa, which corresponds to a 0.66 GPa yield stress is just as likely of a value as the one used above. Using a yield stress of 0.66 GPa would bring the calculated and measured forces into good agreement.

CONCLUSIONS

It has been demonstrated that by adjusting the tunneling current and bias voltage parameters an STM coupled to an electronic balance can be used to perform useful ploughing experiments at nanometer scale dimensions. The ploughs produced showed evidence of instabilities the exact origin of which are not known at this time. These instabilities made comparison to a plasticity analysis more difficult but, by using average values, both the experimentally measured vertical force and the plough's geometry were shown to be in reasonable agreement with the predictions.

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THE ROLES OF KNIFE SHARPNESS AND CUTTING SPEED IN FILM SLITTING

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Introduction

Photographic film and paper are manufactured in wide sheets and then slit to the desired width. It is the aim of the slitting process to produce straight edges while minimizing the amount of debris generated. This goal requires an understanding of the relationship between slitting parameters and the response of the material. Two of the most important parameters are knife sharpness and slitting speed.

Measuring Knife Geometry

If the slitting process is modeled and the optimum knife characteristics are determined, it then becomes necessary to accurately measure the cutting edges of the slitter knives to control the fabrication process. Three techniques for measuring knife sharpness (i.e. knife edge radius) were investigated. The methods involved 1) examining a cross section of the scissor blade in a SEM, 2) creating an impression of the blade in a plastic material and viewing a cross section of the replica with a light microscope and 3) moving a probe with a known geometry across the cutting edge of the blade and calculating the edge radius from the recorded displacement of the probe. The variation in calculated edge radius was about 7% for the three methods. Although each technique provided an accurate and repeatable means to measure knife sharpness, the replication method was judged to be the best due to its simplicity, speed and non-destructive nature.

Film Slitting: Experiments and Evaluation

The dependence of cutting forces and film edge quality on knife sharpness and cutting speed were investigated. A pair of scissors served as a simple analogy to the circular, counter rotating knives used in industry. The scissors and commercial knives have similar sharpnesses, angles between upper and lower blades and spring forces between blades. The force required to close the scissors and cut photographic film was measured using a piezoelectric load cell (Kistler 9251A). Polyethylene terephthalate (PET) and cellulose triacetate based films were cut with blades having 1.9, 4.5 and 8.0 μm edge radii at speeds from 2 in/sec to over 100 in/sec.

Figures 1 and 2 show the cutting force as a function of knife edge radius and slitting speed for the two films. For the sharpest blades (1.9 μm edge radii), the PET film showed a nearly linear decrease in cutting force with cutting rate up to a speed of about 60 in/sec before the force appeared to reach some constant level. Increasing the blade edge radius raised the force at every cutting speed. The acetate film displayed a relatively constant cutting force for cutting speeds less than 100 in/sec when slitting with the sharpest scissor blades. An abrupt decrease in force occurred between 100 and 160 in/sec. As the blade became dull through intentional wear, the force transition happened at a lower speed.

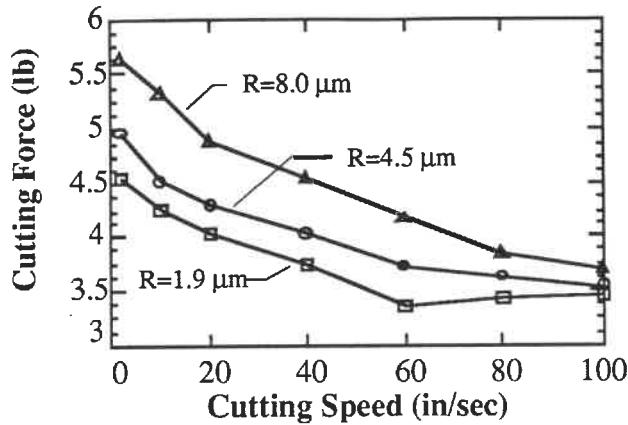


Figure 1. Cutting force vs. cutting speed and blade sharpness for PET film.

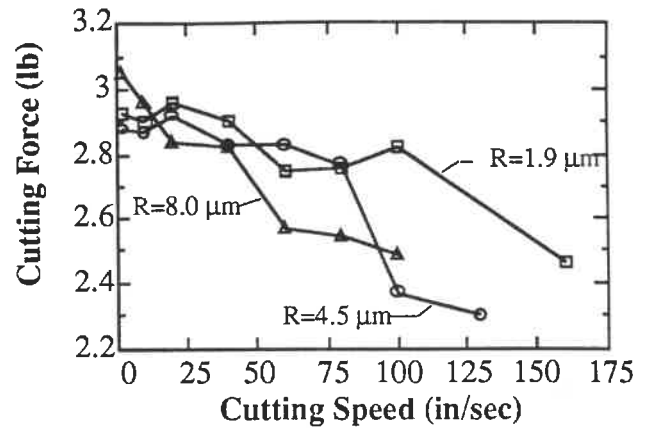


Figure 2. Cutting force vs. cutting speed and blade sharpness for acetate film.

Examination of the slit film edges using an optical microscope provided some understanding of the influence of knife sharpness and slitting speed on debris production and the shearing behavior of the films.

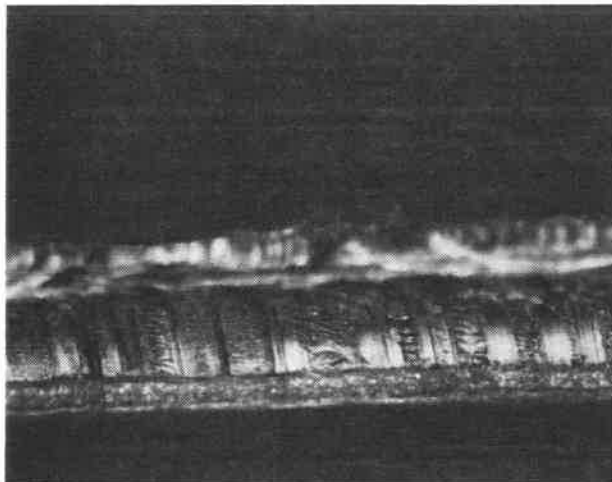


Figure 3. Back face of PET film slit at 2 in/sec with sharp blades, 140X.

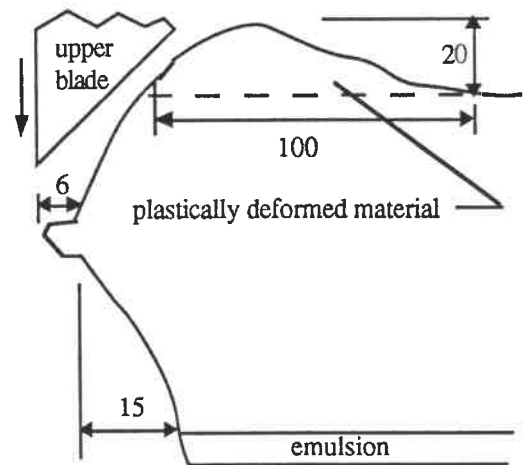


Figure 4. Edge profile sketch, dimensions in μm .

The ductility of the PET based film and the nature of the slit edges were found to be dependent on the slitting speed. Pictured in Figure 3 is a PET film edge slit at 2 in/sec with the sharp ($1.9 \mu\text{m}$ edge radii) blades. The magnification of the edge in Figure 3 is 140X. Figure 3 and the profile sketch in Figure 4 show that the edge that was in contact with the angled face of the blade during cutting was plastically deformed. The upper surface of the edge in Figure 3 is distorted due to this plastic deformation. In contrast, that part of the edge formed by the lower knife is reflective and fairly smooth. Little debris could be found in these edges formed at a low slitting speed. The widths of the rolled and smooth regions that comprised the whole of the cross section of the edges slit at 2 in/sec decreased as the cutting speed increased. At higher speeds, most of the edge showed a region where layers of PET appeared to have been extruded from the base material.

Thread-like debris was present throughout this region. The cross sectional area of the plastically deformed material (indicated in Figure 4) was determined to vary directly with the slitting force shown in Figure 1. It was also found that the slitting force became nearly constant when the area of the deformed region became independent of speed. Furthermore, the film failed in a more vertical plane as the slitting speed increased. The decrease in distortion of the film edge and the increase in debris produced during slitting suggest that the PET film became less ductile as the cutting rate increased.

The type of failure observed for slitting with the sharp blades was also found in the edges produced by blades having 4.5 and 8.0 μm edge radii. Compared to the sharp blades, however, these duller blades damaged the film edge to a greater extent.

In contrast to the ductile PET film, the cellulose triacetate based film was determined to behave in a brittle manner under typical slitting conditions. Shown in Figure 5 is the 70X micrograph of opposing areas on the two edges of an acetate film sample slit at 2 in/sec with the sharp blades. The jagged nature of the surfaces indicates brittle fracture. If the faces are rotated about the black line separating their light colored emulsion surfaces, they would almost fit together (i.e. little debris has been created by slitting process). Preventing a perfect match following this rotation are the crevices that run the lengths of the lower side of the top edge and the upper side of the bottom edge, as shown by Figure 6. This sketch also indicates that no material is raised on the surfaces touching the angled blade faces during slitting. Increasing the slitting speed or blade edge radius resulted in the elimination of the ordered fracture seen in Figure 5. The widths of the crevices in the film edges increased with speed, as did the amount of debris that could be seen in the edges.

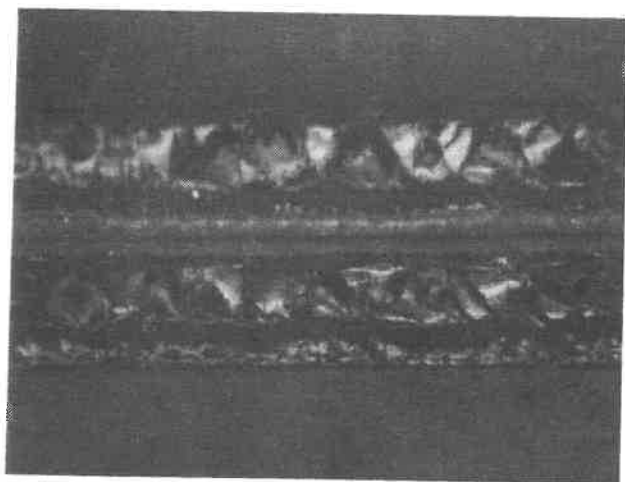


Figure 5. Front and back faces of acetate film slit at 2 in/sec with sharp blades, 70X.

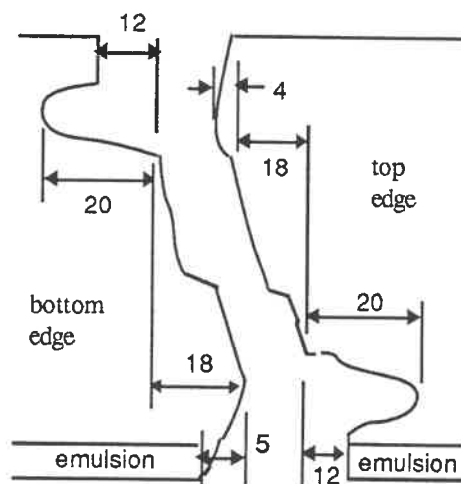


Figure 6. Edge profile sketches, dimensions in μm .

At speeds equal to or greater than those required to produce the force transitions of Figure 2, the edges became relatively square and smooth. The optical microscope showed that the edge angle decreased from about 5° to 3° and the peak to valley distortion across the film thickness decreased from about 30 μm to 15 μm as this force transition occurred. Further examination of the edges using a Talystep surface profilometer showed that the average roughness along the length of the

film edge decreased from about 2.5 μm to 1.0 μm as the cutting force suddenly decreased with cutting speed. These observations suggest that an abrupt change in the edge profile accompanies the rapid decrease in the slitting force. As the shear plane becomes vertical and the roughness of the film edges decreases, the area of the surfaces created by the slitting process decreases, and with it, the energy required to produce these surfaces.

Modeling the Slitting Process

A model of the slitting process was developed to explain the characteristics of the slit film edges and the forces associated with creating them. A quantitative explanation of cutting is available from slip line field solutions to wedge indentation in metals. Slitting is similar to this process and occurs in two phases. During the indentation phase, material along the film surfaces is displaced by pressure from the angled blade faces as the knives penetrate the film. The second phase begins when the transverse tensile stress in the uncut film reaches the yield stress of the material. The remainder of the film is then separated through tensile fracture. Force equations derived from slip line theory relate the blade geometry and the indentation depth to the cutting force. A formula for indentation load as a function of tool geometry and depth was modified to account for the rate dependent properties of the PET film and applied to the slitting problem [1]. A similar equation was used in examining the failure of the more brittle acetate based film at low slitting speeds [2]. Good agreement between the forces predicted by the model and those measured experimentally was obtained. This result suggests that the measured slitting forces correspond to the forces required to indent the films.

Conclusion

Techniques for measuring a cutting tool with less than a 10 μm edge radius are limited to scanning electron microscopy, probing the cutting edge with a tip of known geometry and replication methods. The time consuming and expensive nature of the SEM technique and the high contact stresses associated with the mechanical technique make these methods unattractive for quantifying the many knives of a commercial slitting line. Replication techniques combined with optical microscopy represent a quick, accurate and non-contact mode of knife edge measuring.

The responses of the PET based film and the cellulose triacetate based films to varying knife sharpness and slitting speed were very different. At low slitting speeds, the PET film was mostly ductile and subject to significant plastic deformation. The ductility of the film and cross sectional area of the plastically deformed regions along the film surfaces decreased with increasing cutting rate. The slitting force was determined to vary directly with the amount of plastic deformation. In contrast, the acetate film behaved in a brittle manner under typical slitting conditions. From investigation of the slit edges using an optical microscope, it was also observed that an increase in the edge radii of the blades or the slitting speed generally resulted in a greater amount of debris produced by the process. Furthermore, it was found that slip line field solutions to wedge indentation in metals can be used to model the slitting of polymer films under some conditions.

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STUDY OF FINISHING THE GEAR WHICH IS MADE OF HARDENED STEEL MATERIAL

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Abstract

In this paper, the method used to determine the tool form from the cutting of the hardened gear profile by using theoretical eqs.; the use of heavier cutting conditions for the purpose of decreasing cutting time; findings obtained from cutting with this formed fly tool; measuring the gear by means of a gear tester and a universal measuring machine are described.

Introduction

A gear generating method using a fly tool which is attached to the cutting arbor of a hobbing machine has been used for a long time in various applications such as the tangential feed gear cutting of a worm wheel, the cutting of a ratchet wheel and the serration of an axle as well as test cutting of W-N gears.

The relationship between the gear profile to be cut within the work material and the present tool form will be clarified by the aid of gear profile theory in stead of the numerical trial and error method employed in the conventional fly tool analysis.

The eqs. given below, which determine the form of fly tool decided from the cut gear profiles, have already been derived and have been shown to be correct through experiments made by the authors.(1)

Gear generation using this type of fly tool have none of the irregular contours on the cut gear surface which are inevitable in conventional hobbing.

Furthermore, because this tool shape is simple, it is easy to manufacture, this fly tool method is also useful for reducing the cost of high-precision special rotor profiles of a screw-compressor.(2)

For another example, the use of a cutting edge made of wBN and cBN materials will be successfully applied to the finishing process of the gears which are made of hardened-steel materials.(3)

In this paper, the determination of the tool form from the cutting gear profile by using theoretical eqs., and findings obtained from cutting by this formed fly tool and measuring the accuracy of the gear by means of a gear tester and a universal measuring machine are described.

According to experiments, a very good hardened gear profile has been obtained. It can be reasonably expected that the gear cut by the present formed fly tool will have its accuracy increased by various improvements in its manufacturing process.

Theoretical eqs.

(Determination of the tool form from the cutting gear profile)
The derived eqs. are:

$$\tan \beta \{ (\varepsilon (y \tan \Gamma + z) - y \sec \Gamma) \sin(t + \phi + \theta + \sigma) - (\varepsilon (S - x) \tan \Gamma + x \sec \Gamma) \cos(t + \phi + \theta + \sigma) \} + (\varepsilon (S - x) + g \sec \Gamma) \cos \sigma = 0 \quad (1)$$

$$(y \sin \Gamma + z \cos \Gamma) \cos(\varepsilon \theta + \gamma) + (S - x) \sin(\varepsilon \theta + \gamma) = F = r_a \sin \gamma \quad (2)$$

$$\left. \begin{aligned} X &= (S - x) \cos(\varepsilon \theta + \gamma) - (y \sin \Gamma + z \cos \Gamma) \sin(\varepsilon \theta + \gamma) \\ Z &= y \cos \Gamma - z \sin \Gamma \end{aligned} \right\} \quad (3)$$

Experimental Method

Cutting edge tool points X_i , Z_i and corresponding values of R_i are given by substituting values of θ_i , ϕ_i that satisfy eqs. (1) and (2) simultaneously into eq. (3). (See Fig. 1.)

The formed fly tool used for the experiments was made by WEDM(Wire Electro Discharge Machine), then finally ground for final finishing using these values of X_i and corrected values of Z_i . To obtain the enough negative rake angle on the fly tool(-30.0°) for the purpose of keeping the rigidity of the tool edge, the rake surface of the tool was held up from the center of the tool. (See Fig. 2 and 3.) Then, to obtain the relief angle on the fly tool, the X-axis of the tool was inclined at 16° to the horizontal. Furthermore, in these test cutting, the unipurpose arbor for the fly tool was made and tool was inserted into the two precise inner right-angled surfaces in the square whole which is located in the appointed position.

Fig. 3 shows the tool rad. and offset of the tool face.

Fig. 4 shows the theoretical shape of the tool and machining errors.

Fig. 5 shows the formed fly tool made in this way.

No tangential feed is necessary in this gear cutting method, which is inevitable in the conventional fly tool method.

Table 1 gives the specifications of the cutting gear.

Table 2 gives the cutting conditions.

In this method of gear cutting, both rotational centers of the tool and the workpiece are made to agree with the rotational center of the hob saddle.

However, by using the CNC hobbing machine, the centering process was overcome completely by giving calculated values.

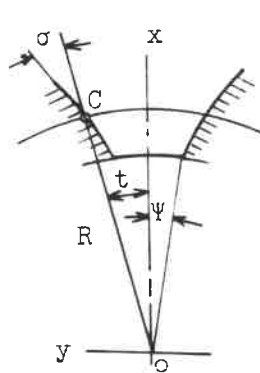


Fig. 1 Involute Tooth Profile

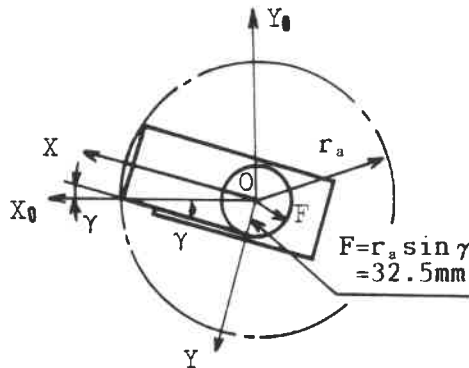


Fig. 2 Relationship between Rake Angle and Offset of Tool Face

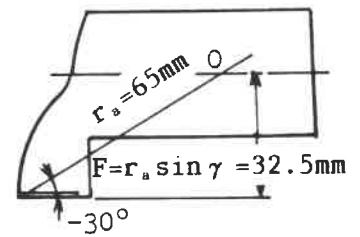


Fig. 3 Tool Rad. and Offset of Tool Face

Nomenclature

C : contact point between gear and tool
 F : $r_a \sin \gamma$ (offset of tool face)
 f : feed rate of gear cutting
 N : number of teeth of the gear
 R : distance between gear rotation center and point C
 r_a : distance between tool rotation center point and point C
 S : distance between both centers of gear and tool rotation
 t : parameter of gear profile
 x } : coordinate axes of the gear
 y } cut by the formed fly tool
 z } (z : axis of rotation of gear)

X_0 } : coordinate axes of the
 Y_0 } formed fly tool
 Z_0 } (Z_0 : axis of rotation of tool)
 X } : coordinates of points on the
 Z } cutting edge tool profile
 β : helix angle of the gear at point C
 Γ : setting angle of the tool
 γ : rake angle of the tool
 ε : cutting ratio
 θ : rotated angle of the tool
 θ_i : See Experiment.
 σ : tangential line angle
 ϕ : parameter of gear profile
 ϕ_i : See Experiment.

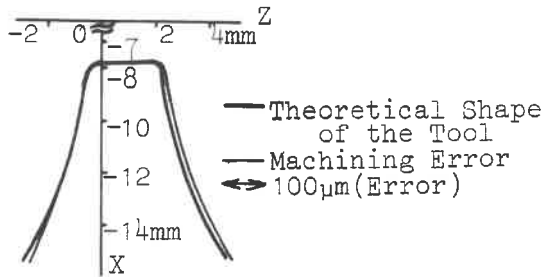


Fig. 4 Theoretical Shape of the Tool and Machining Error

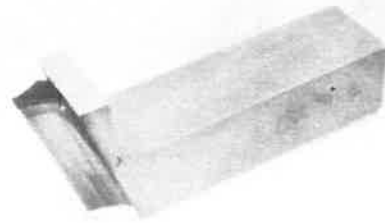


Fig. 5 Formed Fly Tool

TABLE 1 Gear Specifications

Module	mn=3mm
Pressure Angle	20°
Number of Teeth	N=20
Helix Angle(Right-hand)	20°
Face Width	20mm
Pitch Circle Diameter	63.85mm
Material	S45C(=1045)
Hardness(Quenching, Tempering)	H _R C 50

TABLE 2 Cutting Conditions

Setting Angle	18° 35' 48"
Rake Angle	-30°
Offset of Rake Face	32.5mm
Tool Rad.(Max.)	65mm
Revolution	500rpm
Cutting Speed	200m/min
Feed Rate	0.25mm/rev

Results

Fig. 6 shows the gear which was cut by formed fly tool.

Table 3 gives the accuracy of the gear which was measured by means of the gear tester and the universal measuring machine.



Left Right
Fig. 6 Gear Surfaces Manufactured

TABLE 3 Accuracy of the Gear

Single Pitch Error	7µm
Pitch Variation	6µm
Accumulative Pitch Error	16µm
Tooth Profile Error(Right)	11µm
(Left)	13µm
Lead Error(Right)	15µm
(Left)	5µm
Tooth Surface Roughness(Right)	3µmRmax
(Left)	5µmRmax

Discussion

According to experiments, a very good gear profile was obtained by using eqs. (1), (2) and (3).

In these experiments, several improvements were added to obtain a better gear by using severer cutting conditions.

The followings are some experiments of such improvements.

1) In these experiments, the stronger type hobbing machine which has sufficient accuracy, enough rigidity and a bigger diameter cutting arbor(d=90mm) were used.

2) To obtain a more precise fly tool profile, the shape of the fly tool should be designed as shown in Fig. 5.

The actual tool profiles which were used in the previous experiments fitted very well with the theoretical ones.(See Fig. 4.)

But it was difficult to grind the bottom portion exactly because of the limitation of the grinding wheel dia. and shape, because of the interruption between the wheel and the shoulder portions of the tool. So, the shoulder portions of the tool had to be removed in order to easily grind and to get more precise tool manufacturing.

- 3) The tool waist portion was the weakest part. So, in order to avoid this demerit, the equipped portion of the cutting arbor had to be removed and a stronger tool shank against the cutting force was used. (See Fig. 7.)
- 4) Using the hydraulic expansion arbor for work arbor, it was easy to adjust the fly tool to the work material.

So, the method of exchanging the fly tool become easier, if wear or chipping occurred on the fly tool.

After adding these improvements to the previous experiments, a better gear was obtained with severer cutting conditions.

Concerning as manufacturing errors, all items are kept within 3-grade in JIS (Japanese Industrial Standard).

Further, cutting time was decreased because of the heavy cutting conditions.

It can be reasonably expected that the method of gear cutting by present formed fly tool will have its accuracy and its cutting efficiency increased by still more improvements in its manufacturing process.

The followings are some examples of such improvements

1. The tool which was used in the experiments had the relief angle in the X-axis direction. But grinding the tool in the normal direction was better and tool has the relief angle in normal direction, the tool will be remade again by grinding only the rake face, and so the tool should be used for a longer period of time.

2. For the purpose of increasing the cutting speed and cutting efficiency, the tool shape should be designed as shown in Fig. 8, and the pluralization of the tool blade will be necessary.

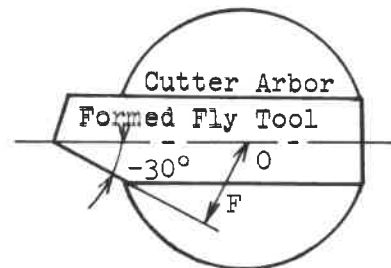
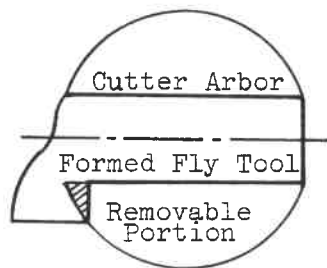


Fig. 7 Improvement of the Cutter Arbor Fig. 8 Improvement of the Tool Shape

Conclusion

The results are summarized as follows.

- 1) Because of the simple tool form, this formed fly tool method can be applied to cut the hardened-gear tooth profile having high accuracy and a very small surface roughness.
- 2) By using the fly tool with sufficient negative rake angle, the stronger type hobbing machine and the special cutting arbor, the severer cutting conditions were obtained and the cutting time was decreased.
- 3) It can be reasonably expected that the gear cut by the present formed fly tool will have its accuracy increased by still more improvements in its manufacturing process.

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Development of a Deep Hole Boring Tool Guided by Laser: Boring of Workpieces with a Thin Wall and an Inclined Prebored Hole

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1 Introduction

A laser-guided BTA tool has been developed to prevent hole deviation, which results in degradation of the quality and a decrease in the yield rate of products. To examine the performance of this tool, fundamental experiments were carried out [1]. Results showed that the tool moved according to computer instructions and that it bored following the target, which was set in front of the tool and moved from side to side. In this experiment, rear cams as actuators remained in the guide-bush.

In this study, the guiding method and apparatus were improved in order to bore such deeper holes (hole length 380mm) that three rear cams could enter the machined hole. The performance of this tool was examined using workpieces with several factors causing the hole deviation (e.g., thin hole wall and inclined prebored hole) [2, 3, 4].

2 Experimental apparatus

An experimental apparatus is shown in Figure 1. A He-Ne laser (power 5mW, 9) with a polarizing plate and two photo-detectors are set in front of the tool. Optical head 8 is composed of a polarizing beam splitter, a corner cube prism and reflecting mirror. The corner cube prism and reflecting mirror are used for measuring tool position and tool inclination, respectively.

The laser beam radiated from the He-Ne laser is deflected on the optical head. The returning beam for measuring tool position reaches photo-detector 11 through the half prism (fixed on the front surface of the laser source). The returning beam for tool inclination does not pass through the half prism, but goes directly to photo-detector 10.

3 Guiding method

Figure 2 shows the setting condition of the tool and guiding strategy in X direction. Z is the rotation axis of the spindle. Z_p axis is parallel to Z with a space of δ . $Z_p - \delta_x$ and $Z_i - \delta_i$ are rectangular coordinates of position and inclination, respectively. The latter is inclined at θ with the former. δ_x and δ_i are the change of each spot on the detectors for measuring position and inclination, respectively. The changes of tool position δ_x , tool inclination $\frac{d\delta_i}{dz_i}$ and the rate of increase of tool position $\frac{\Delta\delta_x}{\Delta z_p}$ are defined as shown in Table 1. Further, raising and lowering motions of the top of the tool are defined as + and -, respectively, when tool inclination is controlled. Further, raising control, noncontrol and lowering control are defined as V_1 , V_2 and V_3 , respectively. The tool is controlled according to the following logic equation [5].

$$V_1 = \overline{U_1} \overline{U_2} + \overline{U_1} U_2 \overline{U_3} \quad (1)$$

$$V_2 = U_1 \overline{U_2} \overline{U_3} + \overline{U_1} U_2 U_3 \quad (2)$$

$$V_3 = U_1 U_2 + U_1 \overline{U_2} U_3 \quad (3)$$

Table 2 and Figure 2 show the demonstration of control. At a hole depth Z_1 , δ_x , $\frac{d\delta_i}{dz_i}$ and $\frac{\Delta\delta_x}{\Delta z_p}$ are U_1 , $\overline{U_2}$ and U_3 , respectively. The logical product $U_1 \overline{U_2} U_3$ is in V_3 of equation (3). Similarly, at the depths of Z_2 , Z_3 , Z_4 and Z_5 , inclinations of tool are controlled like V_2 , V_1 , V_2 and V_3 , respectively.

4 Experimental procedure

Forward cams were used only for setting of the tool. Only rear cams were used for control during machining.

Experiments were carried out with the rotational tool-stationary workpiece system. Rotational speed N is 270rpm and feed f is 0.125mm/rev. Tool diameter is 110mm. Counter-boring of a duralumin (A2017-T4) workpiece with a prebored 108mm dia. hole, was performed. Normal workpieces and one which had a thin wall on $+X$ side were used (Figure 3). A workpiece with an inclined prebored hole meant the normal one which was set in the condition that a prebored hole was inclined toward $+X$ direction at the rate of 100 μ m to a hole depth of 100mm. The cutting fluid was sulfo-chlorinated oil (JIS type 2-13) and was supplied from the laser side.

Four kinds of experiments were carried out. The first experiment was to examine the conditions of the hole deviation in the case of normal deep hole boring. The second was to examine whether the tool corrects for the stated hole deviation. The third was to examine whether the tool can go straight through to the target without shifting toward a thin wall during boring of the thin wall workpiece. The fourth was to examine whether the tool can be guided toward the target without affection of an inclined prebored hole.

5 Experimental results and discussion

Variation of each hole deviation in X and Y directions is shown with a hole depth l in Figure 4.

- [a] **Boring of a normal workpiece (without guiding)** Hole deviated by 74 μ m and 72 μ m toward $+X$ and $+Y$ directions, respectively, at a hole depth of 100mm (Figure 4(a)). It was deduced that this deviation was due to the inclination of the tool occurred during tool setting.
- [b] **Boring of a normal workpiece (with guiding)** In X direction, the hole deviation that occurred in experiment [a] was corrected and decreased to the range of $\pm 30\mu$ m over a hole depth of 340mm (Figure 4(b)). In Y direction, similar to X direction, the hole deviation that occurred in experiment [a] was corrected. But the hole deviation inclined as shown by the thin continuous line. It was deduced that the inclination occurred when the workpiece was set on the table of a numerical control milling machine to measure hole deviation. Hole deviation was measured by an electric micrometer.
- [c] **Boring of a thin wall workpiece** Hole deviation was controlled within the range of $\pm 30\mu$ m over a depth of 360mm in X direction without shifting toward the thin wall (Figure 4(c)). In Y direction, the direction of the hole deviation changed toward $+Y$ direction from a depth of 100mm. But spot change on the detector was within the range of $\pm 15\mu$ m. Therefore, it was clear that the tool was guided toward the target. The reason why the hole deviation toward $+Y$ direction occurred despite the small spot change was thought to be due to the loose clamping of the workpiece during boring.
- [d] **Boring of a workpiece with an inclined prebored hole** Hole deviation was limited to the range of $\pm 30\mu$ m over a hole length 370mm in X direction (Figure 4(d)). It showed that the tool advanced without being affected by an inclined prebored hole.

6 Conclusions

The performance of the laser guided tool was examined using workpieces of three different shapes. The following were cleared for X direction.

- In the case of the boring of a normal workpiece without guiding, the hole deviated by 74 μ m at a hole depth of 100mm.
- In the case of the boring of a normal workpiece with guiding, hole deviation decreased to the range of $\pm 30\mu$ m at a depth of 340mm.
- Even if a thin wall workpiece was bored, the machined hole did not deviate toward the thin wall. Hole deviation did not exceed the range of $\pm 30\mu$ m over a depth of 360mm.

presumably to supply more gold ions to the irradiated region and to remove the microbubbles therefrom, rather than to reduce the diffusion layer.

4. Conclusion

Gold electrical contacts of $\sim 1\mu\text{m}$ height are produced in less than 0.5s for spots and at a laser beam scan speed of more than several hundreds $\mu\text{m/s}$ for lines using an argon laser with an output power of several watts. The diameter of spots and the width of lines are controlled at less than $100\mu\text{m}$ by laser power and cathode potential, as well as by laser beam diameter. Such laser plated spots and lines have a fine morphology, high adhesion to the substrate, and low electrical resistivity.

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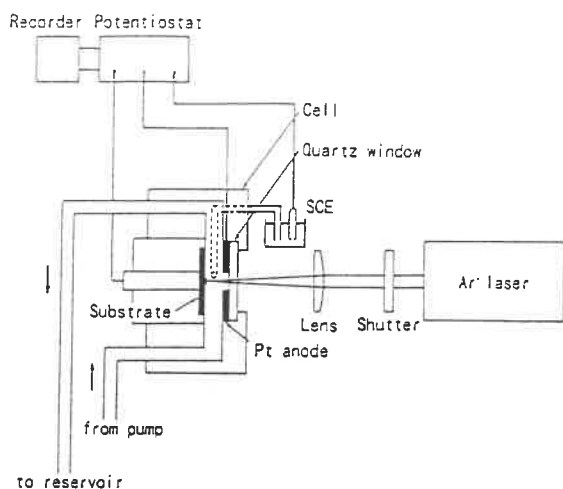


Fig.1 Experimental apparatus for laser plating.

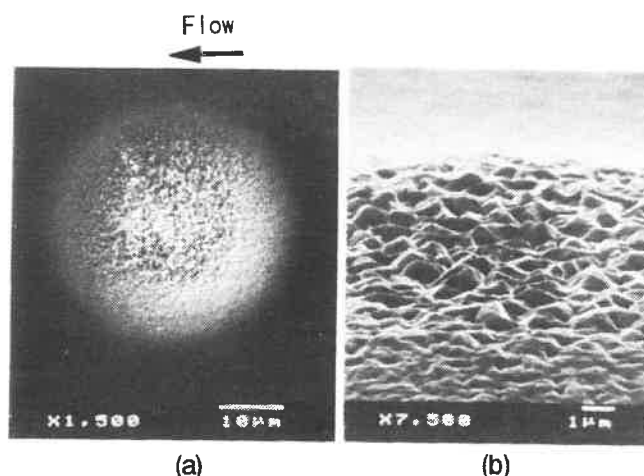


Fig.2 SEM micrographs of a typical spot deposited at 3W, -700mV(SCE), and the flow rate of 1.33m/s for 1s. (a) spot, (b) surface morphology (75° stage tilt)

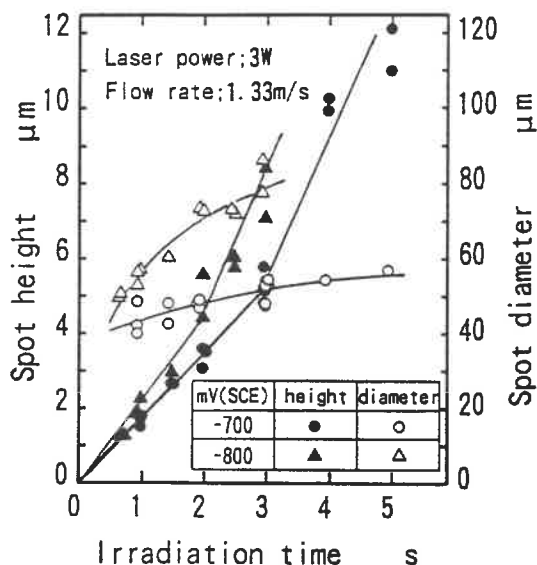


Fig.4 Time dependence of spot height and diameter.

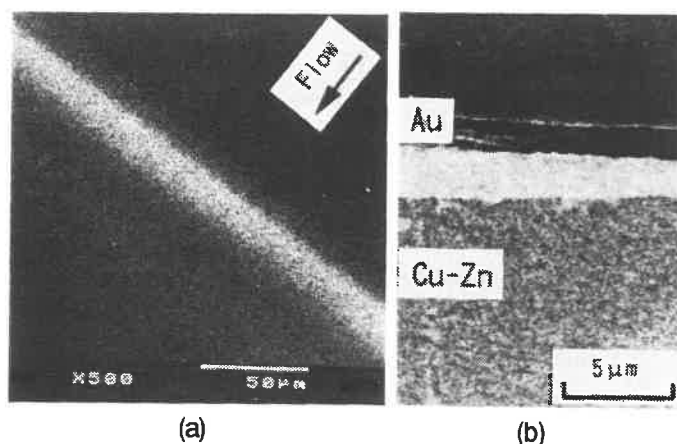


Fig.3 SEM micrographs of a typical line deposited at 4W, -1100mV(SCE), the beam scan speed of 250m/s, and the flow rate of 1.33m/s. (a) line (30° stage tilt), (b) cross section

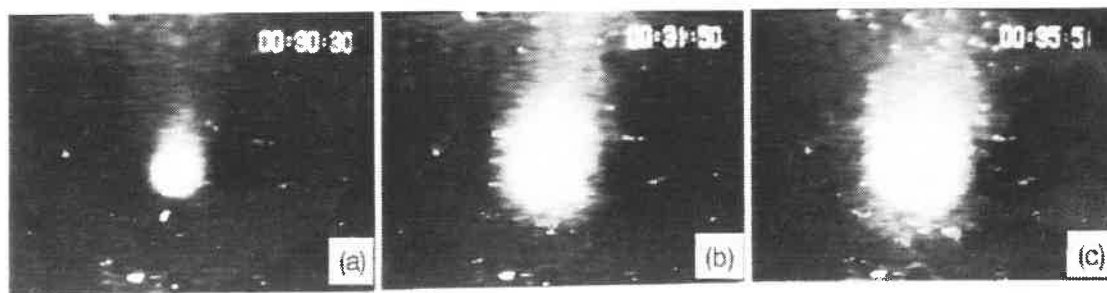


Fig.6 Typical result of in situ observation of laser plating.
(a) $t=0s$ (The instance of laser irradiation), (b) $t=1.2s$, (c) $t=5.2s$

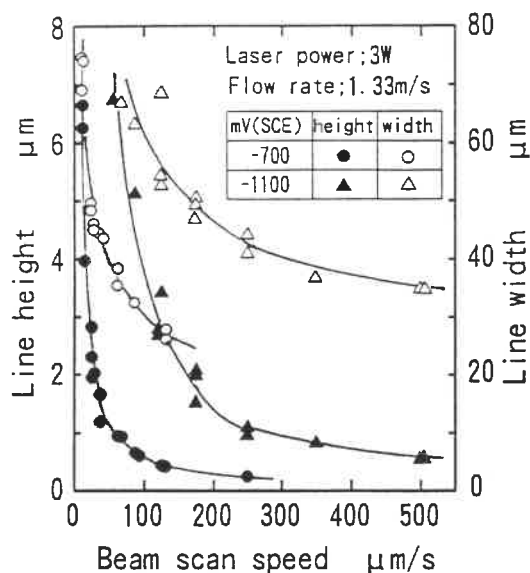


Fig.5 Variation of line height and width with laser beam scan speed.

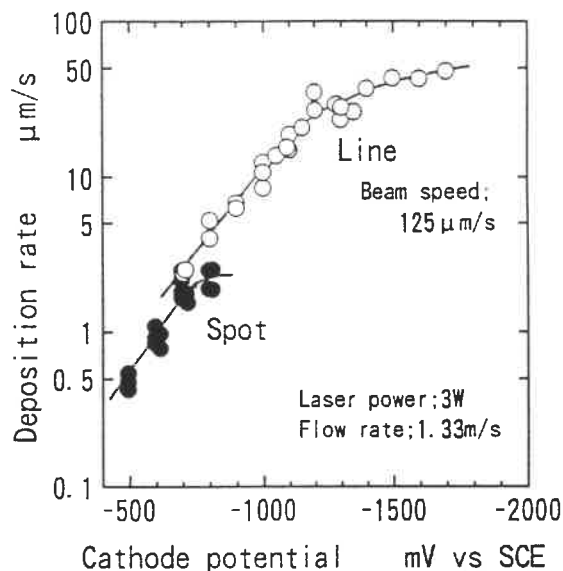


Fig.7 Effect of cathode potential on deposition rate for spot and line plating.

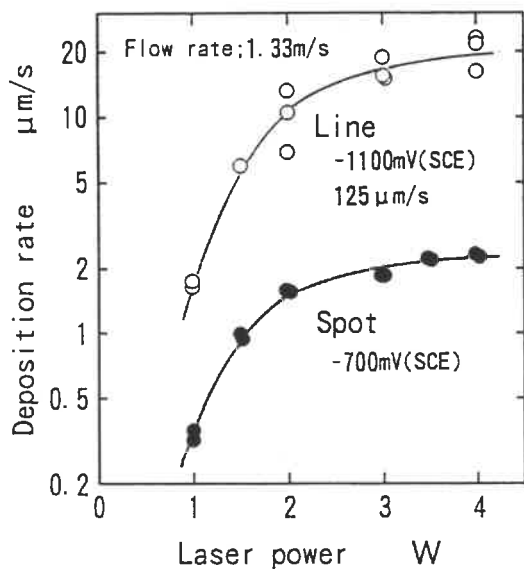


Fig.8 Effect of laser power on deposition rate for spot and line plating.

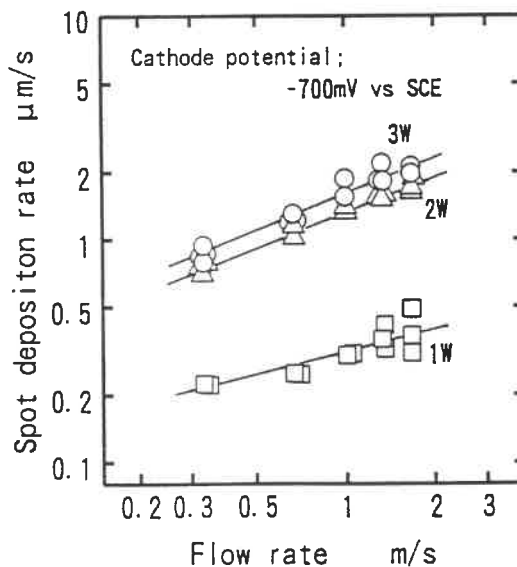


Fig.9 Relation between spot deposition rate and solution flow rate.

OXYGEN ION BEAM ETCHING OF DIAMOND (100)

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1 INTRODUCTION

In recent years, many dry etching techniques such as ion beam etching (IBE), reactive ion etching (RIE), reactive ion beam etching (RIBE) and ion beam assisted chemical etching (IBAE) have been used for fine line engraving in VLSI and opto-electrical devices. Those techniques also can be used for fabricating micro-mechanical parts such as micro-machines and diamond tools. Diamond tools like styli and indenters are usually polished mechanically, using fine diamond power and a lap plate of soft material. In contrast to the mechanical polishing, ion beam etching is more effective in fine etching of hard and brittle materials such as diamond. In dry etching techniques, particularly, RIBE has the advantages of low radiation damage, high etching rate, good selectivity and so on.

This paper reports the dependences of the etching rate of diamond on the ion energy and the ion incidence angle under "reactive" oxygen ion beams, the surface roughness evaluation of diamond chips (100) after oxygen ion beam etching and so forth.

2 EXPERIMENTAL APPRATUS AND PROCEDURE

The (100) face of single crystal diamond chips were used as specimens. The specimens were etched using an ion beam etching apparatus with a Kaufman-type ion source [1] shown in Figure.1.

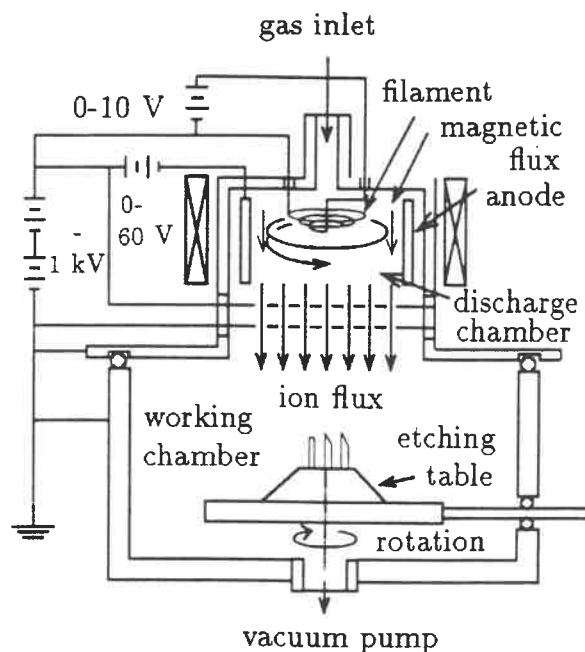


Figure 1: Ion beam etching apparatus with a Kaufman-type ion source.

Ions are generated in the discharge chamber by low energy electron bombardment of the operating gas. They are then extracted and accelerated through the extraction grid and enter the working chamber, and eventually strike the specimen. Kaufman-type ion sources produce several mA/cm² ion flux at energies in the range of 500-1000 eV, and are available up to about 8 cm diameter. As for the present experimental apparatus, a Kaufman-type ion source is not suitable owing to erosion of the tungsten filament by the reactive (oxygen) species. Therefore, we desire to carry out experiment using an ion beam etching apparatus with an ECR (electron cyclotron resonance)-type ion source [2], operated without filament.

The schematic of the specimen holder to fix diamond chips during ion beam etching is shown in Figure.2.

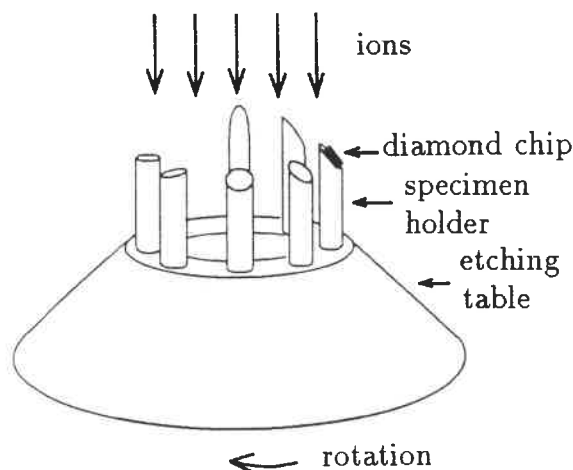


Figure 2: Schematic of the specimen holder and the etching table.

As shown in the figure, diamond chips are fixed on the (brss-made) specimen holder having top angle from 0° to 70°, and put in the hole of (aluminum-made) etching table. As the etching rate of diamond is fairly lower than that of the etching table, aluminum atoms sputtered from the etch-

ing table redeposit on the diamond surface. Thereby, the etching table has enough height in order to prevent redeposition. Moreover, so as to unify the current density of incidence ions, the etching table is rotated by a direct current motor.

3 EXPERIMENTAL RESULTS AND DISCUSSION

The most important characteristics in etching or machining of diamond are the etching rate V , which depends on the following matters [3].

- Incidence ion species.
- Ion energy E (keV).
- Ion incidence angle ($^\circ$).
- Ion current density (mA/cm²).
- Crystal face and direction of the specimen.
- The temperature of the specimen during the ion beam etching.

The etching rate V dependence of the (100) face of single crystal diamond on the above mentioned parameters using oxygen ions were investigated. Those data of the angular dependence of etching rate V , however, are most important for the etching of micro-mechanical parts made of diamond.

The dependence of the etching rate V of diamond (100) on the ion energy E is shown in Figure.3. As shown in the figure, the etching rate V increases with increasing the ion energy, but reaches a saturation level at an ion energies above 900 eV. When the ion energy exceeds a certain value, physical sputtering is easily occurred, therefore we think that the etching rate V decreases. Moreover, compared with ion beam etching

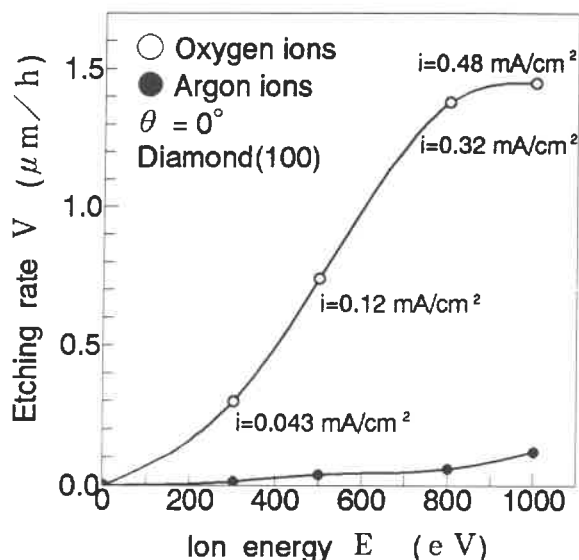


Figure 3: Dependence of the etching rate of diamond (100) on the ion energy.

using “inert” argon ions, the etching rate for oxygen ions at the ion energy of 1000 eV is about fifteen times larger than that for argon ions. Because chemical reaction of diamond and oxygen ions and ($C + O^+ \rightarrow CO$, CO_2) exceeds physical reaction for what is called sputtering.

The dependence of the etching rate of diamond (100) on the ion incidence angle of oxygen ions is shown in Figure 4. At the ion energy of 500 eV, the etching rate V decreases gradually with increasing the ion incidence angle θ which is defined the angle between the surface normal and incidence ions whereas the etching rate V at the ion energy of 1000 eV increases with increasing the ion incidence angle, and reaches a maximum. The maximum occurs at the ion incidence angle of approximately 30° , and then may decrease rapidly with increasing the ion incidence angle. The reason why the etching rate V decreases as the incidence angle increase, incidence ions are merely reflected at the target surface, without sputtering any atoms. In order to analyse the

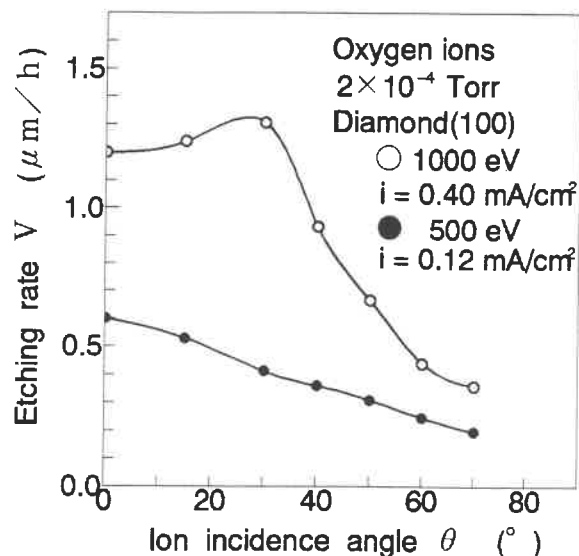


Figure 4: Dependence of the etching rate of diamond (100) on the ion incidence angle of oxygen ions.

details of reactive etching reaction mechanism, we must be investigated further.

4 SURFACE ROUGHNESS EVALUATION

The surface roughness evaluation of diamond (100) face after oxygen ion beam etching examined using an Atomic Force Microscope (AFM) [4]. The AFM has a cantilever with the very soft spring constant ($0.1 \sim 1$ N/m) [5] and can measure ultra-low load of 10^{-9} N level. As the AFM need not an electric conductivity of the specimen, it can measure the surface roughness of an all objects in principle. Thus an AFM thinks fit to measure the surface roughness of diamond.

4.1 RESULTS

The measured area of a diamond chips by AFM is $15 \times 15 \mu m^2$ and the measured points are 512×512 points. Figure 5 shows an AFM image of the diamond chip before

the ion beam etching with a surface roughness R_z of 6.98 nm. Figure.6 shows an AFM image of diamond chips after 90 minutes etching under oxygen ion beams with a surface roughness R_z of 0.83 nm. Compared with before oxygen ion beam etching, a surface roughness R_z is an about eighth part smaller than that of the pre-finished one.

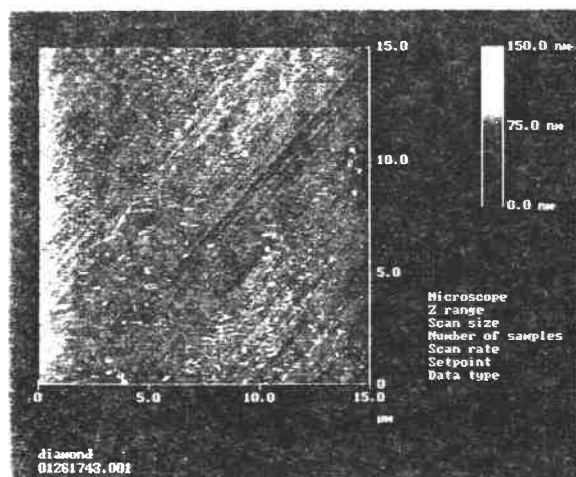


Figure 5: AFM image of a diamond chip before ion beam etching.

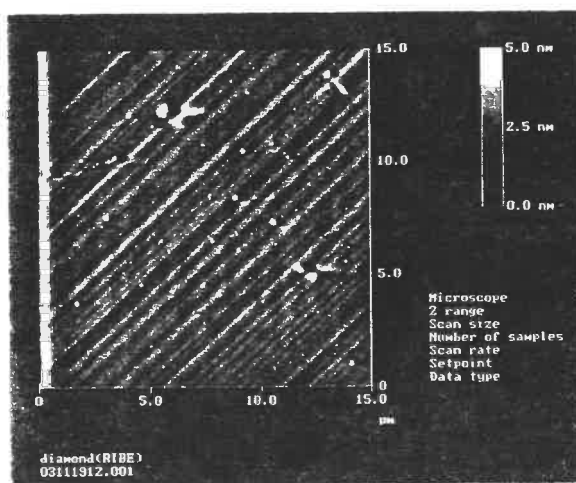


Figure 6: AFM image of the diamond chip after 90 minutes etching under oxygen ion beams.

5 CONCLUSIONS

Applying oxygen ion beam etching method for diamond specimens, we obtained the following conclusions.

- For oxygen ion beam etching includes both physical sputtering and chemical reactive etching, the etching rate is about ten times larger than that of ion beam etching using argon ions which causes only physical sputtering.
- Surface roughness of diamond measured using AFM is decreased by oxygen ion beam etching method.

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Newly Developed Grinding and Lapping Tools Applying Anodic Spark Deposition of Fine Abrasives

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Summary

The present paper describes the manufacture of new abrasive tools applying anodic spark deposition (ASD) and the application of them to the grinding and lapping processes. The newly developed ASD process is able to make fine abrasives rigidly fix and uniformly deposit on the substrate because of the formation of a firm anodic layer. The abrasive wheel wear test were carried out to estimate the grinding ability of glass materials. It was found that the material removal depth depends on the kind of abrasives and the contact pressure of the wheel, and that the grinding ratio also depends on the contact pressure. The good surface integrity were accomplished under the optimum grinding condition. It was concluded that the newly developed abrasive tools applying ASD process had sufficient grinding ability for brittle materials.

Background

The use of fine abrasives is a common strategy to realize the good surface integrity in the grinding process of brittle materials. But, on the other hand, it seems very difficult to fabricate a homogeneous and tough grinding tool composed of fine abrasives of less than 1 μm in diameter because of low bonding strength, which results from the small adhesive interface between the grains [1]. In order to overcome this difficulty, a new grinding or lapping technology has been developed by applying anodic spark deposition of fine abrasives [2].

Anodic spark deposition (ASD) is a relatively new method of forming ceramic coatings by means of a spark [3]. The ceramic material is deposited on the positive electrode in an electrolytic bath, which comprises an aqueous solution and ceramic fine particles dispersed in it. According to this method, various coatings composed of fine abrasive grains and anodic compounds can be successfully formed on some substrate metals regardless of their shapes without the bonding agents. The prepared fine abrasive coatings have a high potentiality of grinding and lapping tools because they have such advantages as good adhesion to the substrate and comparatively porous structure, which plays the important role of the chip pockets.

Anodic Spark Deposition

ASD process is an unconventional process for forming ceramic coatings on conductive substrates. As the name implies, the ceramic material is deposited on the positive electrode of an electrolytic cell by a mechanism involving a spark [4].

Anodizing, a similar electrochemical process which is the precursor of anodic spark deposition, ordinarily occurs at potentials of the order of tens of volts and current densities of the order of microamperes per square centimeter. Coating thickness varies with voltage, and if electrolyte and deposition conditions are selected properly, electrically insulating coatings (barrier film) form on some materials. The dielectric strength of these coatings varies with the anode materials and the electrolyte; however, when the applied field exceeds the maximum dielectric strength of the anodized coatings, dielectric breakdown of the film occurs. Sparking reactions due to dielectric

breakdown result in the deposition of material from the electrolyte.

A newly developed ASD process in this study is basically different from the method described above in the use of aqueous solutions dispersing ceramic fine particles such as aluminate and silicate [5]. Figure 1 shows a schema of the experimental arrangement. In this process, negatively charged particles in an alkaline solution move to the anode and deposit themselves on it accompanied with normal ASD process simultaneously. Figure 2 is SEM photograph of silicate and aluminate coating by the new ASD process. Coatings are generally porous and amorphous structure, and also strongly adherent to the substrate, which results from the generation of a mixed layer of anodic compounds and ceramic particles at a interface between the substrate and the coating. Various coatings can be obtained by the new ASD process as given in Table 1.

Grinding Test and Results

In order to investigate the grinding ability of the newly developed ASD coatings, grinding tests of glass materials were carried out using a ASD abrasive wheel. Figure 3 shows the experimental apparatus of the ASD abrasive wheel grinding test. The ASD abrasive wheel is composed of an aluminum alloy core (50 mm in diameter, 12 mm in width) and the ASD abrasive tape adhering around it, which thickness is about 50 μm including the aluminum foil substrate. The aluminum foil was examined for the substrates because it was best metal for good adhesion and quite useful for any application. The work piece, lime glass material, was reciprocated on the ASD abrasive wheel at constant contact pressure and at 42 mm/s in average speed. As the wheel was rotated at an angle of 0.9 degree every reciprocation of the work piece, flesh grains always worked on the work piece. No grinding fluid was used in this experiment.

Figure 4 shows the dependence of the removal depth of the ground glass (R_p) and the wear depth of the silicate and aluminate ASD abrasive tape (R_v) on the number of reciprocations. Both the R_p and the R_v increase in proportion to the reciprocation at the same rate independent of abrasive tapes. Figure 5 shows the relationship between the number of reciprocations and the surface roughness of the ground glass (R_a). The R_a decreases to about 0.5 μm by a reciprocation increase. This suggests that the surface roughness of the ground glass seems to reflect that of abrasive tapes. In order to improve the surface roughness, it is necessary to investigate the ASD abrasive coatings and the process, and also to develop how to handle them and how to grind or lap.

Figure 6 and Figure 7 show the dependence of the grinding ratio of the silicate and aluminate abrasive tape (R_p/R_v) and the surface roughness of the ground glass (R_a) on the contact pressure respectively. In the case of using the aluminate abrasive tapes, the R_p/R_v and the R_a do not depend on the contact pressure under the present experimental conditions. On the other hand, in the case of the silicate tapes, the R_p/R_v increases by a contact pressure increase, but the R_a is not varied in this condition. Figure 8 is the relationship between the wear depth of the abrasive tapes and the removal depth of the ground glass. The grinding ability under 400-800 reciprocation times is higher than that of others. This indicates that it is necessary to dress the ASD abrasive coating.

Conclusions

According to the newly developed anodic spark deposition process, various ceramic coatings composed of fine abrasives can be successfully formed on the aluminum substrate without bonding agents because of the production of an anodic layer. From experiments of ASD abrasive wheel grinding, it was found that ASD abrasive tapes have sufficient grinding ability for glass materials. It was concluded that ASD abrasive tools have great potentiality in a field of precision machining. Improvement of the surface roughness and grinding ratio is the subject for a future study.

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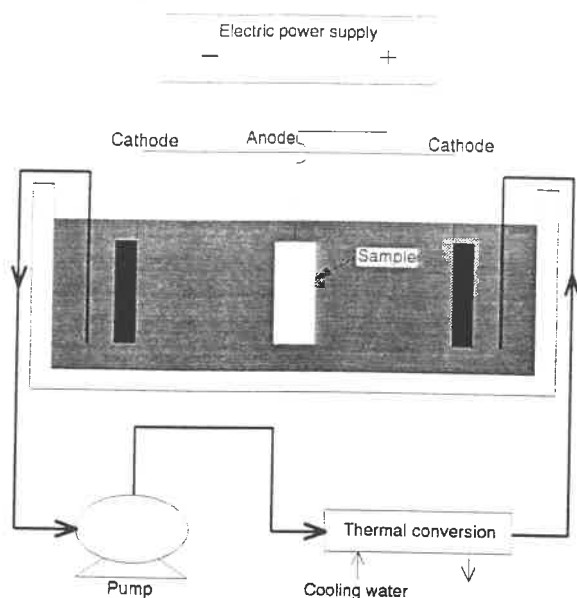
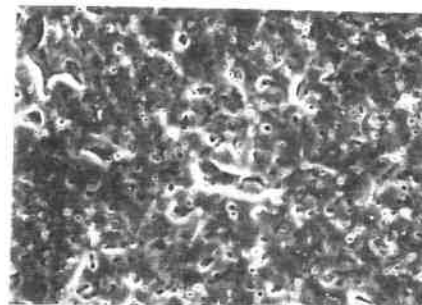
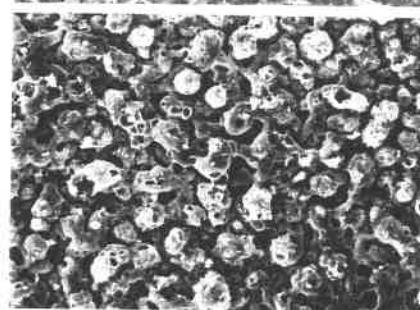


Figure 1. Schema of Experimental Arrangement of New ASD Process

(A) Aluminate Coating



(B) Silicate Coating



100 μm

Figure 2. SEM Photograph of Silicate and Aluminate Coating

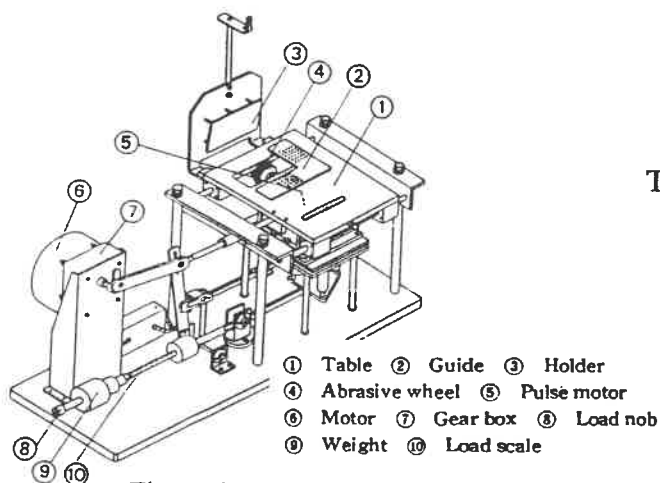


Figure 3. Experimental Apparatus of ASD Abrasive Wheel Grinding Test

Table 1. Various Coatings by New ASD Process

Element of abrasive film (name)	Suitable thickness (μm)	Surface roughness μmRa	Hardness (pencil)
Al_2O_3 (AO)	5~8	0.7	5
$\alpha\text{-Al}_2\text{O}_3$ (AA)	10~20	0.9	7
$\text{Cr}_2\text{O}_3\cdot\text{Al}_2\text{O}_3$ (CA)	8~12	0.7	7
$\text{Al}_2\text{O}_3\cdot\text{Mo}$ (AM)	5~8	0.6	4
SiO_2 (SI)	10~20	2.1	3
ZrO_2 (ZR)	10~20	1.1	7

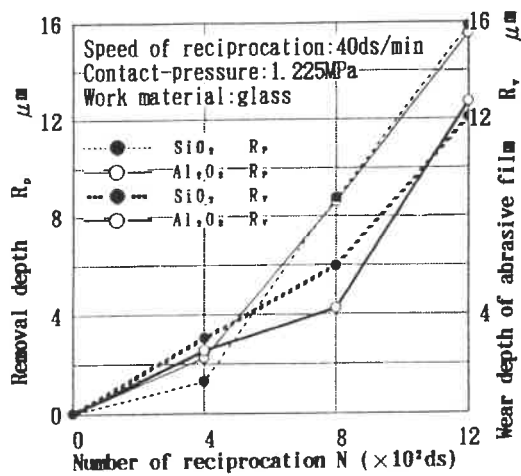


Figure 4. Dependence of Removal Depth of Ground Glass and Wear Depth of Silicate and Aluminate ASD Abrasive Tape on Number of Reciprocations

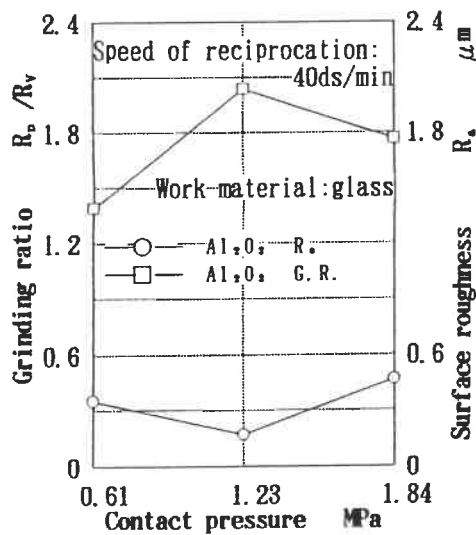


Figure 6. Dependence of Grinding Ratio of Aluminate Abrasive Tape and Surface Roughness of Ground Glass on Contact Pressure

Figure 8. Relationship between Wear Depth of Abrasive Tapes and Removal Depth of Ground Glass

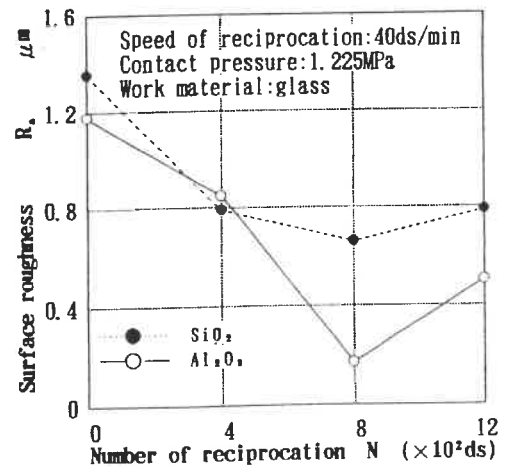


Figure 5. Relationship between Number of Reciprocations and Surface Roughness of Ground Glass

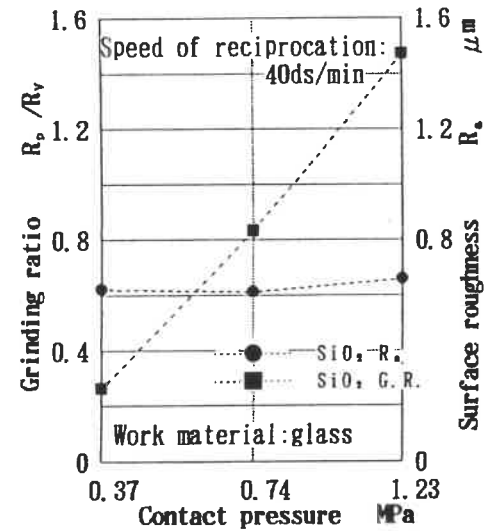
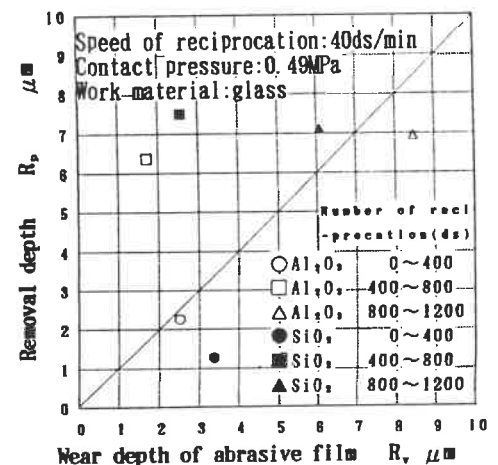


Figure 7. Dependence of Grinding Ratio of Silicate Abrasive Tape and Surface Roughness of Ground Glass on Contact Pressures



HIGH EFFICIENCY ION BEAM ETCHING FOR SMOOTH SURFACE FORMATION OF POLYCRYSTALLINE MATERIAL

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1. Introduction

Recently, in the field of precision tool and mechanical part machining, a technique to satisfy both form accuracy of microns and surface roughness of sub-microns is required. Coining using a punch is very useful for mass production, but conventional methods such as electrical discharge machining (EDM) cannot machine the fine punch to coin a thrust plate of spiral groove bearing, because the punch is required high precision. Ion beam etching shows promise of machining hard and brittle materials with high precision, however, it is necessary to improve the etching efficiency.

It is well-known that the surface roughness of etched surface increases as the etching proceeds. It is considered that this increase is closely connected with the difference in etch rates among crystal grains of polycrystalline materials. It has been reported[1] that uniformly changing the ion incident angle to the normal direction of specimen surface suppresses this increase in roughness.

In this work, in order to reconcile both high accuracy and efficiency, we have investigated the dependence of surface roughness and etch rate on the ion incident angle, using a TRIM computer code[2] simulation technique as a supplement. As a result, a precise punch was obtained using the ion beam etching method taking into account angular dependence.

2. Experimental

For all experiments, a high-speed tool steel (JIS:SKH54, HRC 60), with a grain size of 2-3 μm after heat treatment, was used. A photosensitive polymer of 15 μm thickness was prepared as the etching mask on the specimen surface which had been prefinished by mechanical polishing with a surface roughness of 0.06 μm R_{max} . Etching conditions are shown in Table 1. The specimens were etched by argon ions using a Kaufman-type ion source with a rotating machining table.

Table 1 Etching conditions

Ion energy (E)	700 eV
Ion current density (I)	0.1-0.4 mA/cm ²
Ion incident angles (θ)	30°, 60°, 75°

The etched depth and surface roughness of the specimens were measured by a mechanical stylus instrument. Scanning electron microscope(SEM) and Auger electron spectroscopy(AES) were also used to analyze the etched surface.

3. Results and discussion

Fig.1 shows the relationship between surface roughness and etched depth at various incident angles(θ). The surface roughness of the specimens increases with an increase in the etched depth, asymptotically approaching a certain limit. The surface roughness at $\theta=75^\circ$ becomes constant when it reaches $0.3 \mu\text{m}$ R_{max} . Fig.2 shows the etch rates at various incident angles. The etch rates of α -Fe were calculated, because the main element of the specimens was Fe. The calculated results are in fair agreement with the experimental results, but the peak of the calculated values is shifted to a slightly higher angle. The shift may be caused by the differences in binding energies and densities of steel and α -Fe.

The etch rate at $\theta=75^\circ$ is lower than that at $\theta=60^\circ$, though the etching itself is suitable for high accuracy machining. Moreover, it can be calculated that the etch rate at $\theta=75^\circ$ is approximately one-third the rate of $\theta=60^\circ$, because the ion current density is in proportion to $\cos\theta$. Therefore, it was expected that control of the incident angle would result in high accuracy and efficiency. Fig.3 shows the experimental results which confirmed our expectations. In the first step, the specimen was etched for high efficiency at $\theta=60^\circ$ until the etching reached a depth of $3.2 \mu\text{m}$. In the second step, the angle was adjusted to 75° for high accuracy until the total etched depth reached $5.0 \mu\text{m}$. It can be seen that a surface roughness of $0.3 \mu\text{m}$ R_{max} can be obtained efficiently at a depth of $5.0 \mu\text{m}$. This method reduces the etching time to 280min., which is approximately half

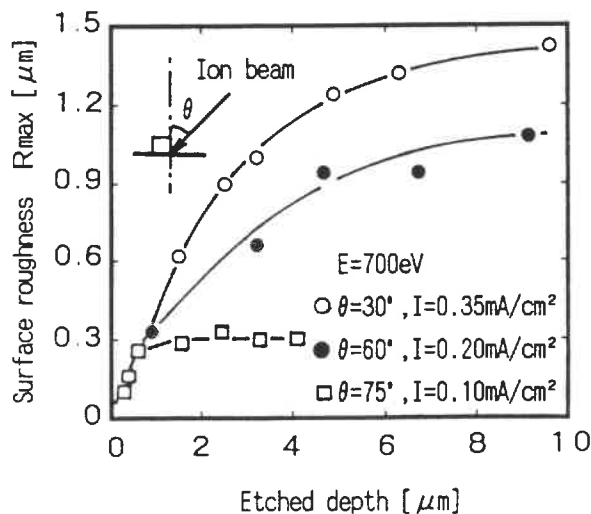


Fig.1 Surface roughness as a function of etched depth

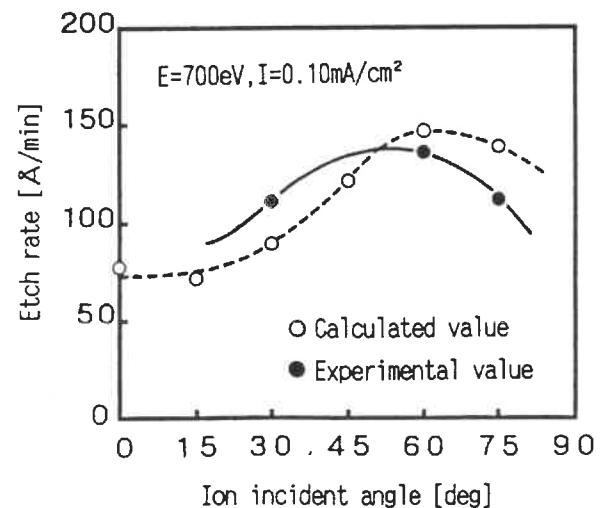


Fig.2 Etch rate as a function of ion incident angle

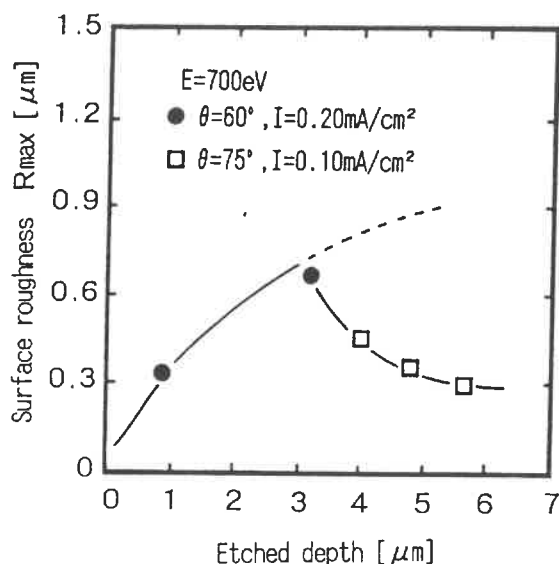


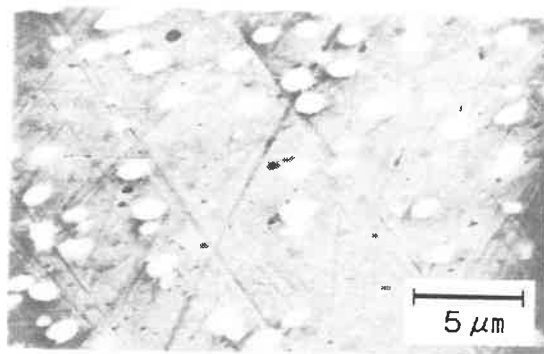
Fig.3 Improvement in surface roughness by adjusting the incident angle from 60° to 75°

that at a fixed angle of 75° .

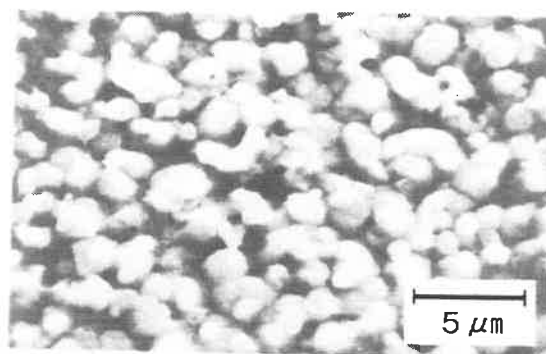
Fig.4 shows SEM images of the specimen surface demonstrating the changes in surface roughness. There are remarkable irregularities in the etched surface at $\theta=60^{\circ}$ (b). In contrast, a smooth surface can be obtained by changing the angle from $\theta=60^{\circ}$ to 75° (c). We believed that the increase in roughness was caused by the difference in etch rates between the base metal and precipitated carbides. Thus, AES was performed to investigate the change in carbide distribution. However, no significant differences were discovered before and after etching. We are convinced that this increase in roughness by etching is due to selective sputtering of the disordered grain boundary of the base metal.

Furthermore, it is postulated that convergence of surface roughness by etching at $\theta=75^{\circ}$ depends on the interruption of ions at the hollow grain boundary. Fig.5 schematically illustrates this concept of the etching mechanism.

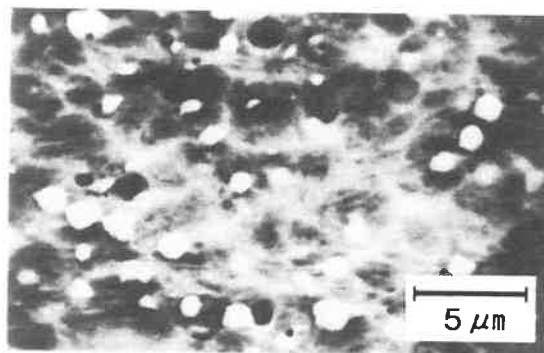
This method was applied to machining a punch that had a fine spiral groove on the surface (accuracy; groove depth $5\mu\text{m}\pm0.2\mu\text{m}$, surface roughness $0.3\mu\text{m Rmax}$).



(a) As mechanical polished
 $0.06\mu\text{m Rmax}$



(b) Etched depth: $3.2\mu\text{m}$
 $0.66\mu\text{m Rmax}$



(c) Etched depth: $5.0\mu\text{m}$
 $0.30\mu\text{m Rmax}$

Fig.4 SEM images of the surface for each etched depth

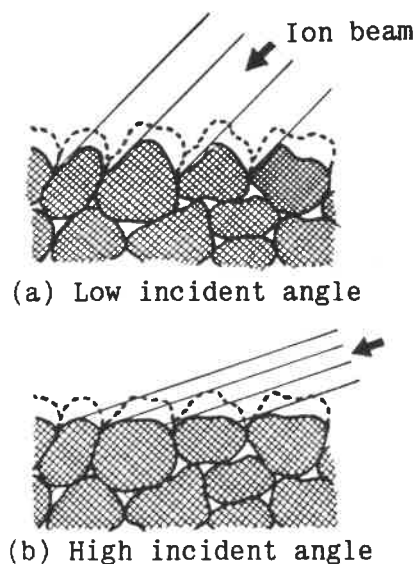


Fig.5 Schematic diagram of polycrystalline material etching

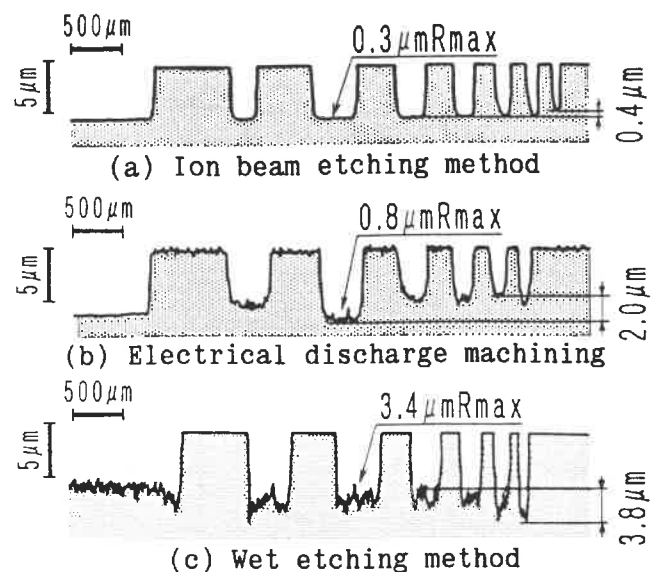


Fig.6 Comparison of form accuracy of punches machined by various methods

Fig.6 shows the form accuracies obtained by this method, EDM, and the wet etching method. It was found that the form accuracy of this method could not be obtained using conventional methods. It is considered that the increase in roughness was caused by the accumulation of spark marks in the case of EDM, while in the wet etching method, it was caused by the difference in etch rates between the base metal and carbides.

4. Conclusion

The angular dependence of surface roughness and etch rate was investigated. The surface roughness of an etched surface approaches a certain limit asymptotically, and the convergent value of the roughness decreases with an increasing incident angle. The etch rate decreases substantially at an incident angle approaching 90° . Highly efficient and accurate etching can be achieved by controlling the ion incident angle, in which adjustment of the angle can reduce etching time and improve roughness. By applying this method, a coining punch having a fine spiral grooved surface can be obtained. By comparing form accuracy to conventional methods, it has been confirmed that the highest accuracy can be obtained using ion beam etching.

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HIGH EFFICIENT ELECTRICAL DISCHARGE MACHINING OF TITANIUM ALLOY

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1. Introduction

Titanium alloy has many excellent properties such as high specific strength, high heat resistance and high anti-corrosion, and is widely used in many fields; for example in the aerospace industry, ocean development industry, nuclear power industry, medical industry and so on. However, titanium alloy is one of the most difficult-to-cut materials because of the large tool wear caused by its high chemical activity and its low thermal conductivity, which leads to an extreme high cutting temperature.

On the other hand, electrical discharge machining (EDM) is generally used for machining of difficult-to-cut materials, since the workpiece is removed by heat and pressure produced in a discharge phenomenon between the electrode and the workpiece, regardless of the mechanical or thermal properties of the material. However, titanium alloy is said to be difficult to machine even in EDM because of the large electrode wear¹⁾.

The factors which make titanium alloy difficult to EDM are experimentally investigated, analyzing the shape of crater generated by a single discharge, the pile up time of the workpiece, the waveform of successive discharge current, the shape of debris and so on. Then the conditions for high efficient EDM of titanium alloy are recommended.

2. Experimental procedures

Experiments are performed with a ram type NC electrical discharge machine equipped with a single pulse discharge device. The workpieces used are titanium alloy which contains 4% aluminium and 6% vanadium, pure titanium and alloy tool steel SKD11. Graphite and copper rods are used as the electrode. The electrical conditions are set as follows; discharge current $I_p=4-60A$, discharge duration $\tau_p=6-1000\mu s$, duty factor D.F.=5-50%.

3. Analysis of a crater generated by a single pulse discharge

Fig.1 shows some craters generated by a single pulse discharge on titanium alloy and alloy tool steel SKD11 under the condition $I_p=40A$, $\tau_p=40\mu s$. In this figure, Gr(+) and Gr(-) represent the electrode polarity, for example, Gr(+) means that the polarity of the graphite electrode is plus and that of the workpiece is minus. As can be seen in the figure, many wrinkles are observed on the surface of the titanium alloy, especially in the case of Gr(+). On the other hand, no wrinkle is observed in the case of SKD11. These wrinkles are generated because of the volume shrinkage of slowly cooling melted material. Since titanium alloy has lower thermal conductivity and higher melting point compared with steel, it is liable to be in the above mentioned state and generate wrinkles.

Fig.2 shows the ratio of pile up height to crater depth for various electrode conditions. As shown in the figure, the ratio is largest in the case of Cu(+), especially the ratio for titanium alloy which reached 0.62. On the other hand, it is relatively low in the case of Gr(-), and that for SKD11 was 0.08. The ratio for titanium alloy is higher than that for SKD11 in all cases.

Fig.3 shows the time for pile up of the workpiece t_s . This is measured by using a short-circuit between the electrode and the workpiece. The chain line $t_s=\tau_p$ shows the case that the pile up of the workpiece is over at the same time as the end of discharge

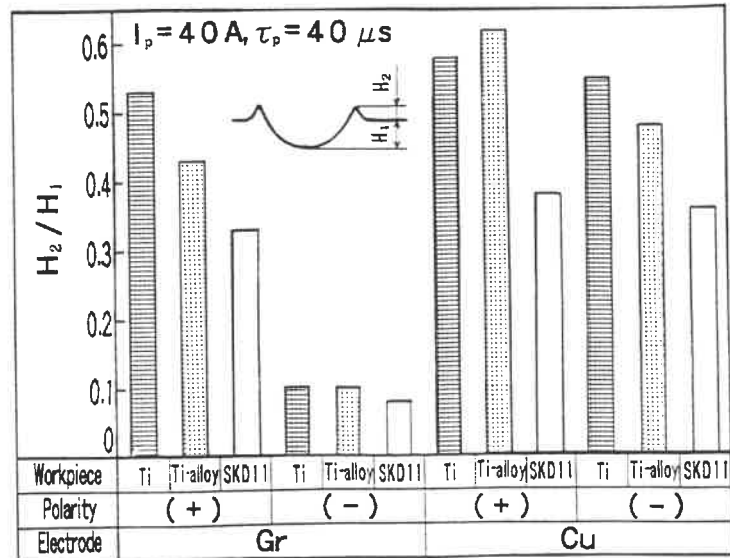
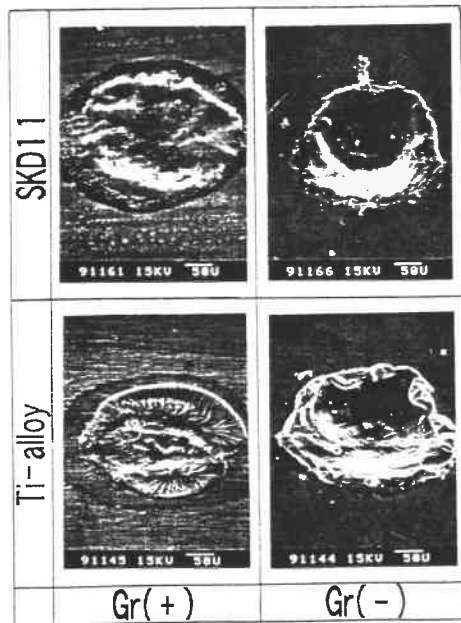


Fig.2 Ratio of pile up height to crater depth

Fig.1 SEM photographs of craters generated by a single pulse discharge

duration. As shown in the figure, however, the pile up phenomenon takes place even after the discharge ($t_s > \tau_p$). The pile up time t_s increases with an increase of discharge duration τ_p . Moreover, the pile up time for titanium alloy is longer than that for SKD11.

Fig.4 shows the relationship between the removal volume by a single pulse discharge V_R and the product of thermal conductivity and melting point $\theta_m \cdot \lambda$. It is generally said that the larger the product $\theta_m \cdot \lambda$ is, the easier EDM is. As shown in the figure, the removal volume V_R increases with an increase of $\theta_m \cdot \lambda$, which agrees with the conventional theory²⁾. It is expected from these results in a single pulse discharge experiment that titanium alloy can be efficiently machined by EDM.

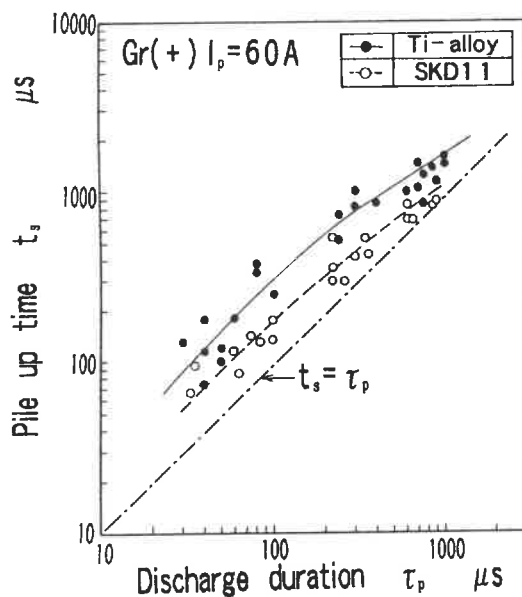


Fig.3 Relationship between pile up time and discharge duration

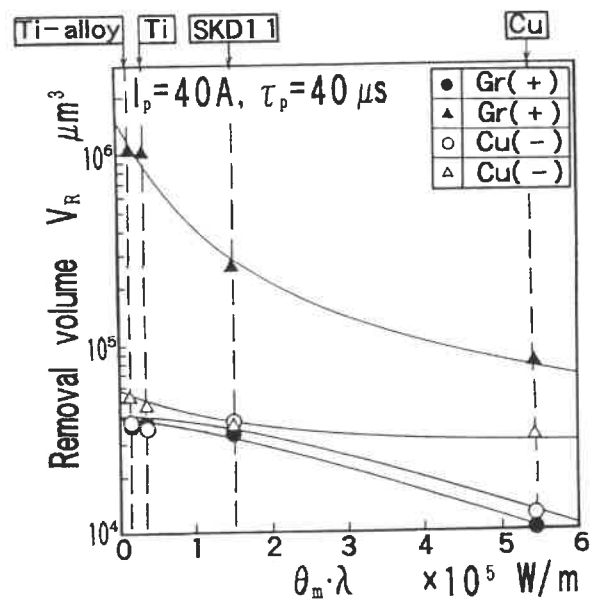


Fig.4 Relationship between removal volume and the product $\theta_m \cdot \lambda$

4. Results of conventional electrical discharge machining

Fig.5 shows the variations in metal removal rate R with the mean discharge energy per time p . As can be seen from Fig.5(a), metal removal rate increases with an increase of energy, it is especially high in the case of the electrode condition Gr(-) and Cu(+). However, as shown in Fig.5(b), the metal removal rates for titanium alloy and pure titanium are less than half of that for SKD11. It indicates that the efficiency of machining titanium alloy is very low, contrary to the experimental results in a single pulse discharge.

Fig.6 shows the waveforms of discharge voltage and discharge current in electrical discharge machining of alloy tool steel SKD11 and titanium alloy. The duty factor which is the ratio of discharge duration to machining time is set at 50% for both cases. However there is a significant difference between the two cases. In the case of SKD11, the duty factor is approximately kept at 50%, while it is much lower than the setting value in the case of titanium alloy. This is due to short-circuit and unstable discharge because of slow but large pile up of titanium alloy, as pointed out in the single pulse discharge experiment.

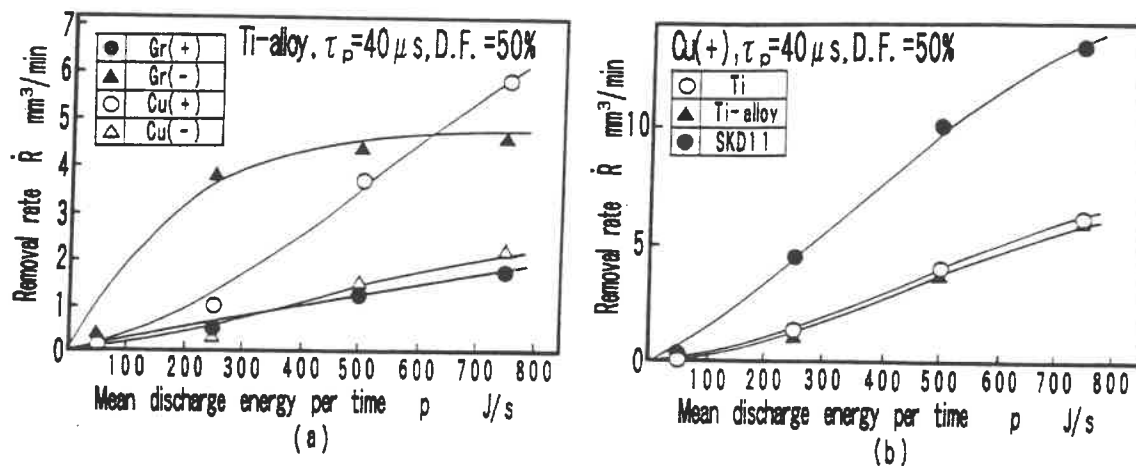


Fig.5 Variations in metal removal rate R with the mean discharge energy per time

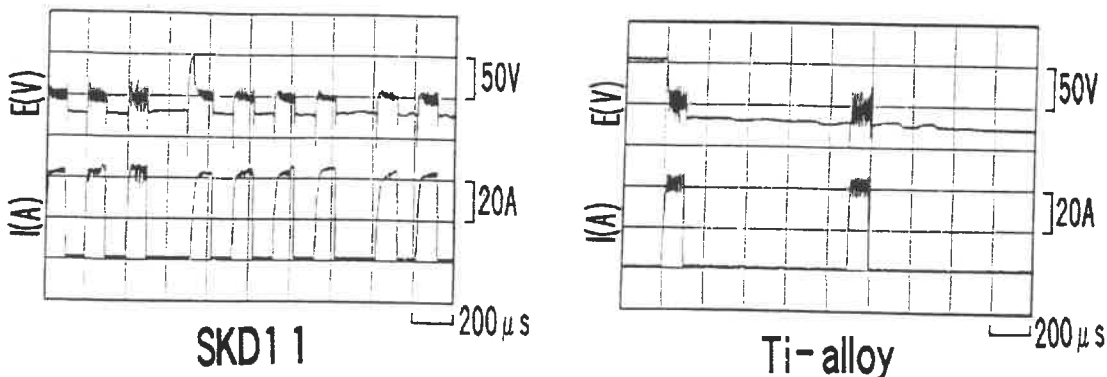


Fig.6 Waveforms of discharge voltage and discharge current ($I_p=40A$, $\tau_p=100 \mu s$, D.F.=50%)

5. Recommendations for high efficient electrical discharge machining

From the above considerations, it is necessary to have enough resting time to pile up for stable and efficient electrical discharge machining.

Fig.7 shows the variations in the metal removal rate with the discharge duration for various duty factor in electrical discharge machining of titanium alloy. The metal removal rate generally increases with an increase of duty factor for steel. As can be seen from the figure, however, the proper duty factor exists which maximizes the metal removal rate for various discharge durations in the case of titanium alloy. On the other hand, the

electrode wear is lower for lower duty factor. Therefore by choosing the proper duty factor, high efficiency and low electrode wear electrical discharge machining of titanium alloy is possible.

Fig.8 shows the variations in the metal removal rate and the electrode wear with the duty factor under the conditions, discharge current $I_p=60A$, discharge duration $\tau_p=300 \mu s$ and the reverse polarity machining Gr(+). As shown in the figure, the metal removal rate increases and the electrode wear decreases with a decrease of duty factor. The electrode wear reaches to 2.4% under the duty factor 10%. Moreover, 0.5% electrode wear is attained under the condition discharge current $I_p=4A$, discharge duration $\tau_p=100 \mu s$ and the copper electrode reverse polarity Cu(+).

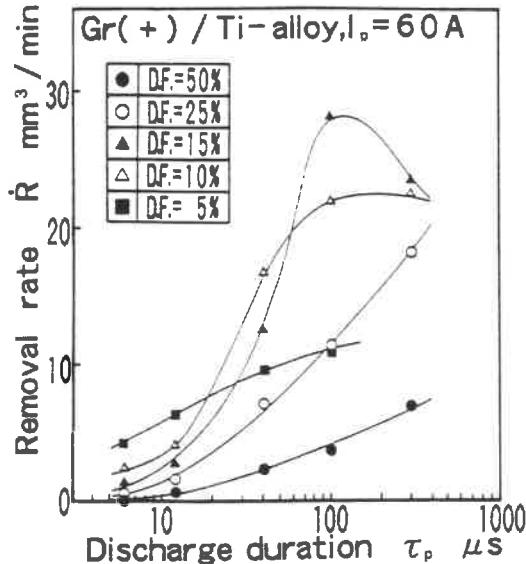


Fig.7 Variations in metal removal rate for various duty factor

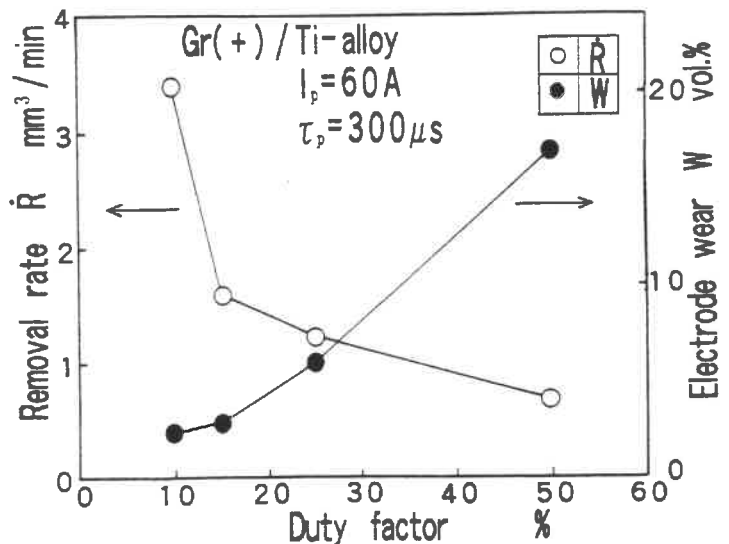


Fig.8 Variations in metal removal rate and electrode wear with duty factor

6. Conclusions

- (1) The pile up generated by a single discharge in titanium alloy is higher than that in steel, and the pile up time is also longer.
- (2) The removal volume of crater generated by a single discharge in titanium alloy is larger than that in steel, which agrees with the conventional theory that the low product of melting point and thermal conductivity of the workpiece leads to the high machinability.
- (3) The metal removal rate is lower and the electrode wear is larger for titanium alloy under the conventional EDM condition as compared with that for steel, since the frequent short-circuits result in the instability of machining.
- (4) High efficiency and low electrode wear electrical discharge machining of titanium alloy is possible by decreasing the duty factor, since the resting duration is enough for pile up of the workpiece and the machining becomes stable.

Acknowledgements

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Using a constant force profiler to investigate the dynamics of mechanical contact at the stylus probe/specimen interface.

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Introduction.

The constant force profiler is a hybrid instrument conceived to bridge the gap between stylus profilers and atomic force microscopes. There is a growing body of evidence¹ that existing stylus techniques may offer certain advantages over AFM for large area metrology. Existing stylus dimensions are more suitable for large artifacts and the styli contact with enough force to "plow" through surface dust and contamination. AFMs, with their micro styli and ultra-low force operation, are preferred instruments for micro-roughness measurements, but suffer from artifacts that are currently subject to some speculation. An operational problem with existing stylus instruments is one of profiling speed. Since most stylus measuring heads have high moments of inertia, and low stiffness ($<35 \text{ N m}^{-1}$ to comply with international standards), the dynamic forces of profiling limit traverse speeds to a few micrometers per second in general. Combining the favorable aspects of AFM probes and stylus instrument designs, we have built a hybrid profiler using ISO standard profiler styli with long range traverse capability at speeds up to 100 times faster than conventional stylus systems.

A block diagram of a recently built second-generation low force stylus profiler^{2,3} is shown in figure 1. Briefly, This instrument utilizes a compensated PZT stack and control system to maintain a constant deflection (and thus force) on a diamond machined glass cantilever of dimension $8 \times 1.5 \times 0.1 \text{ mm}$. This cantilever deflection is monitored by a custom built high-resolution capacitance gage with a noise equivalent capacitance of approximately $1 \times 10^{-18} \text{ F Hz}^{-1/2}$ for a noise equivalent displacement of $3 \times 10^{-12} \text{ m Hz}^{-1/2}$. The specimen is translated underneath the force probe on a ultra-precision slideway effectively referencing motion of the specimen carriage to an optical flat. Linear traverse is 20 mm. The specimen carriage position is monitored by a laser interferometer system with a resolution of 39 nm per count. Quadrature pulses from the interferometer are used to control data acquisition and clocking of the profile. Repeatability has been assessed by direct subtraction of consecutive raw data sets. For these tests, a 740 nm step height standard was chosen as representative of a worst-case specimen (an impulse function is really the worst, but they don't exist in nature). After subtraction, the peak to peak residual is 3 nm. In addition to this high degree of repeatability, speeds in excess of 0.5 mm s^{-1} have been demonstrated while profiling rough specimens. This speed is some 100 times faster than a conventional stylus instrument. Shown in Figure 1 is a block diagram of the profiler system⁴. The force probes for this system are calibrated using a precision force balance⁵ which is, in turn, calibrated with NIST traceable dead weights.

As the stylus loads are reduced and the bandwidth response increases, it is apparent that the dynamics of the probe/specimen contact can play a significant role in the fidelity of the surface profile. Consequently, in this paper we assess the influence of sample materials on the transfer function response of our profiling system.

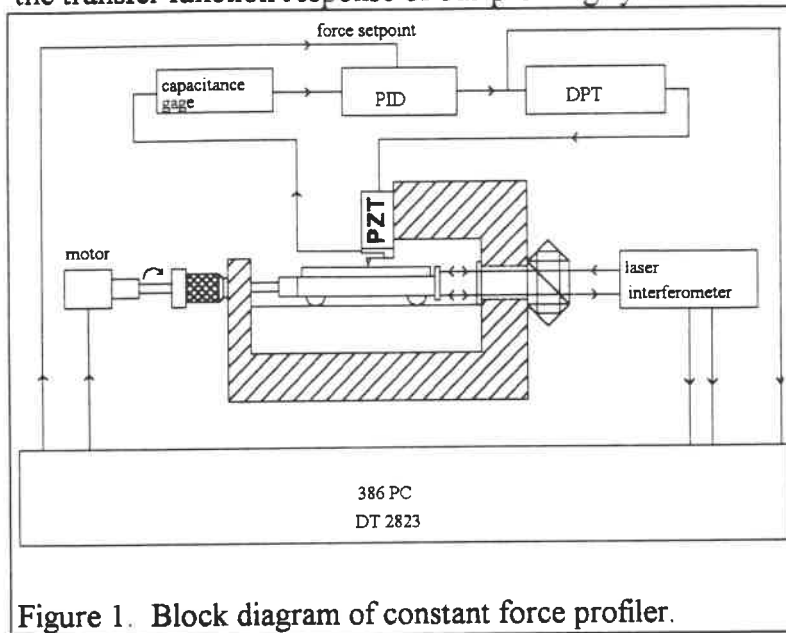


Figure 1. Block diagram of constant force profiler.

Dynamics of contact (theory).

A lumped parameter model of a cantilever force probe in contact with both Voigt and Maxwell type solids is shown in figure 2. The cantilever probe has been modeled as a second order spring mass damper system of effective mass, m_e , stiffness, k_c , and damping force, d_c . The effective mass and stiffness can be found using Rayleighs method and simple beam

theory respectively⁶ while damping, being mainly associated with squeeze film effects in air, must be calculated using numerical methods⁷. The interface has also been modeled as having a constant stiffness, k_m , and a rate dependent damping force, d_m . Again the stiffness at a given nominal load can be derived from Hertzian contact equations and may be considered constant for small perturbations. However, damping is more difficult to predict and will be a combination of interfacial and internal⁸ dissipative forces.

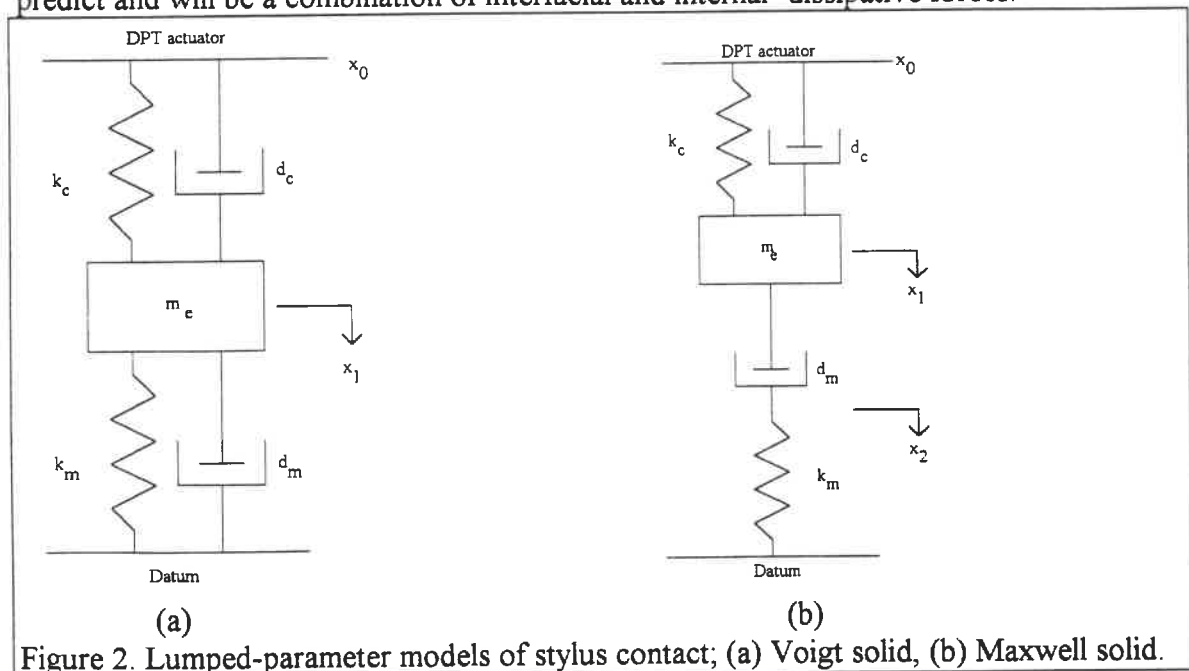


Figure 2. Lumped-parameter models of stylus contact; (a) Voigt solid, (b) Maxwell solid.

To measure the frequency response of our system, a swept sine is applied to the actuator to produce a displacement, $x_o(j\omega)$, and the subsequent deflection of the beam, $x_1 - x_o$, is monitored using a Hewlett Packard 35665A signal analyzer. From the model for the Voigt solid, figure 2(a), the predicted frequency response is

$$H(j\omega) = \frac{x_1(j\omega) - x_o(j\omega)}{x_o(j\omega)} = 1 - \frac{x_1(j\omega)}{x_o(j\omega)} = 1 - \frac{\frac{k_c}{k_c + k_m} + j\left(\frac{\omega d_c}{k_c + k_m}\right)}{1 - \frac{\omega^2}{\omega_n^2} + j\left(2\zeta \frac{\omega}{\omega_n}\right)} \quad (1)$$

where

$$\zeta = \frac{b_c + b_m}{2\sqrt{m_e(\lambda_c + \lambda_m)}}, \omega_n = \sqrt{\frac{\lambda_c + \lambda_m}{m_e}} \quad (2)$$

For a Maxwell solid modeled in figure 2(b) the transfer function is given by

$$H(j\omega) = 1 - \frac{x_1(j\omega)}{x_o(j\omega)} = 1 - \frac{-d_c d_m \omega^2 + k_c k_m + j(d_c k_m \omega + d_m k_c \omega)}{-m_e k_m \omega^2 - d_c d_m \omega^2 + k_c k_m + j(m_e d_m \omega^3 + d_m k_m \omega + d_m k_c \omega + d_c k_m \omega)} \quad (3)$$

The first transfer function, eq. (1), is characteristic of a damped second order system with a dc attenuation of the PZT displacement across the force sensor of magnitude $k_m / (k_c + k_m)$. The frequency response of the Maxwell solid, eq. (3) is more characteristic of a third order high pass filter.

Dynamics of contact (experimental results).

A 2 μm radius diamond stylus tip was used with a static force set to approximately 4 μN and dynamic force variations of approximately 300 nN applied over a frequency range from 10 to 700 Hz. For comparison, two smooth surfaces were measured: Float glass and PMMA from a compact disk substrate. The specimen's surface roughness was determined using a microscope interferometer, and in both cases was between 3 and 5 nm RMS. Several runs were recorded at random locations on each surface, with no ascertainable variations noted for each specimen.

Differences between the two materials can clearly be seen (figures 3 and 4). It was observed that the peak magnification has been reduced by approximately 5 dB from that of glass to the PMMA which also shows a marked attenuation between 10 and 200 Hz.

In a second experiment the viscoelastic Maxwell solid, Silly Putty™, was probed. Viscoelastic polymers similar to this are useful to the extrusion honing process when loaded with abrasive. A 10 μm tip was used with identical forces applied over a frequency range of 1 to 10 Hz. As predicted by equation (3) the response is characteristic of a mechanical high-pass filter, (figure 5).

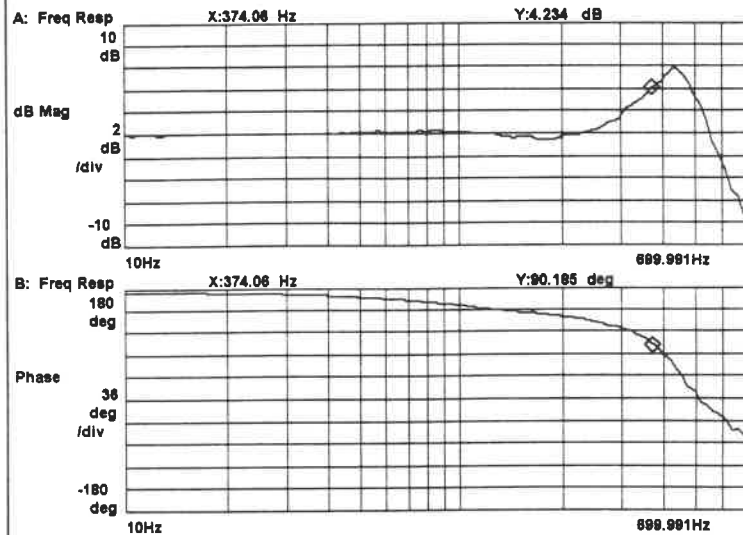


Figure 3. Servo response for contact with glass specimen.

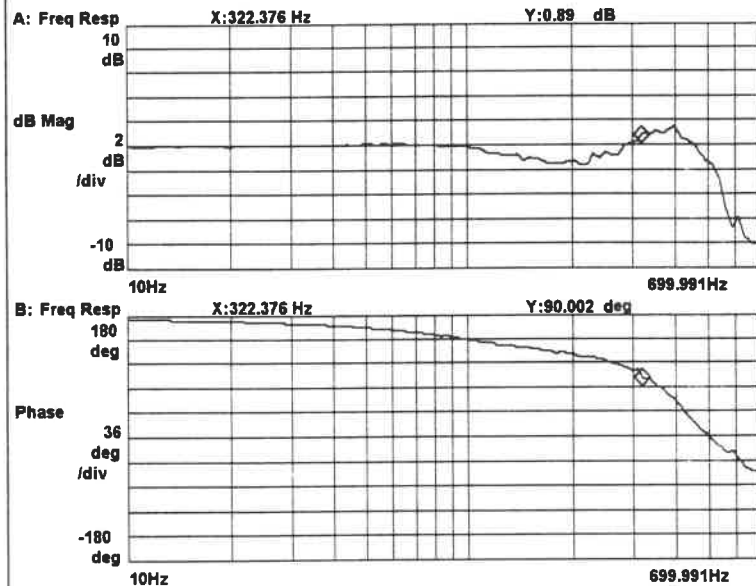


Figure 4. Servo response for contact with PMMA specimen.

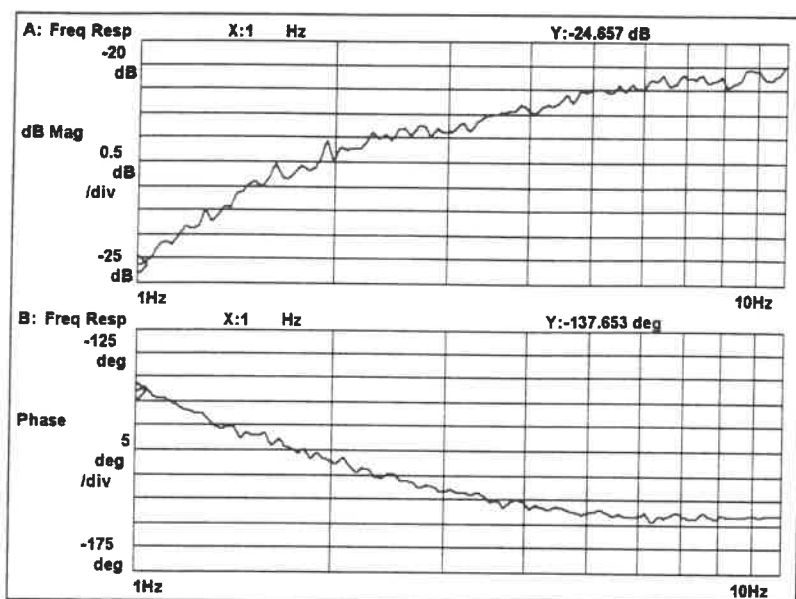


Figure 5. Servo response for contact with putty specimen.

complicate servo-controller tuning. For which specimen is the controller response to be adjusted? How smooth need this specimen be?

- The probe/specimen interaction can determine the mechanical response of viscous materials whose relaxation times are within the linear region of controller response.
- Theoretical models have been developed that may be used to extract quantitative information relating the stiffness and damping characteristic of microcontacts for a given probe/surface interface.
- Voigt solids with relatively short relaxation times can exhibit an effect on the rate-dependent "D" term of a PID controller.

With regards to this last statement, A different controller strategy may be in order for force modulation studies. While for normal profiling and imaging, a flat servo response is desirable, a peaked response resembling an underdamped second-order system is desired for maximum contrast of internally damped Voigt solids. Under such conditions, the phase shift of the servo controller at a frequency near the maximum bandwidth may be monitored. This phase shift may serve as a contrast signal during profiling/imaging. This contrast signal will change with the dynamics of the tip/specimen interface. Alternatively, control system bandwidths on the order of a few KHz would make deconvolution of the surface interaction component much easier because linear responses could then be assumed for the PZT actuator and force sensor in the range from DC to several hundred Hz where most of the mechanical interactions at the probe tip should take place.

Discussion

While it is far too early to draw any sweeping conclusions from this work, the following general statements may be made regarding the dynamics of micro-level "point" contact.

- Tip/specimen interactions are contained within the control loop of the constant force profiler and repulsive-mode AFMs.
- This interaction can

¹ J.M. Bennett, V. Ealings, K. Kjoller, *Applied Optics*, **32**(19), 3442, (1993).

² L.P. Howard and S.T. Smith, *Rev. Sci. Instrum.*, **63**(10), 4289, (1992).

³ L.P. Howard and S.T. Smith, *Rev. Sci. Instrum.*, at review.(1993).

⁴ L.P. Howard and S.T. Smith, *Proceedings IPES 7*, 260, (1993).

⁵ S.T. Smith and L.P. Howard, *Rev. Sci. Instrum.*, at press., (1993).

⁶ L.P. Howard and S.T. Smith, *Rev. Sci. Instrum.*, **63**(10), 4289, (1992).

⁷ Y. Xu and S.T. Smith, *Proceedings of ASPE*, Seattle, November 7, (1993).

⁸C.A. Wert, *J. Appl. Phys.*, **60**(6), 1888, (1986).

Improved Boundary Evaluation of an Indentation by Scanning Tunnelling Microscope data.

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In previous work (1) the boundary of a microhardness indentation was evaluated. The 200×200 array of data $z = f(x, y)$ directly obtained from the STM was represented in a raster bilevel display. The level z is reckoned from the bottom of the indentation. As z increases, some pixels appear which are not connected to those of the indentation. They stay separated from the indentation up to a level z^* , at which some of them become confluent with it. From the centre of the indentation we applied a filling algorithm which captures all the 4-connected pixels. We take the boundary to be situated at the level z^* . This method however suffers from a drawback. Because the STM probe cannot be quite perpendicular to the stage, the measurement of the indentation area is sensitive to specimen orientation. Moreover, this measurement cannot be carried out above the level z^* .

We are now presenting a modified method which is free from these limitations. The STM topographic data are arranged in an increasing order, starting from zero. The first set of data, corresponding to the lowest level, is represented in a raster bilevel display. A filling algorithm is applied, starting from any one pixel. Two files are used, one (called *Inside*) to save filled pixels, corresponding to the inside of the indentation; the other (called *Outside*) to save all the rest. The Inside file is displayed, data of the next level is added to it, and the filling algorithm is applied starting from the same pixel. Filled pixels are saved again in the Inside file, and all the rest are saved in the Outside file. The process is repeated up to the highest level. For each level the areas that lie inside and outside the indentation can be simply found by counting pixels saved in the corresponding files. Inside and outside volumes are given by the cumulate of the same files, starting from the lowest level up to the current one. The proposed procedure is completely automatic. A more refined method applies the filling algorithm both inside and outside the indentation, saving the corresponding pixels in the above-mentioned files. The remaining pixels are left on the display and data from the next level is added to them. For an optimal filling of the outside region we apply the algorithm to all the points of the edges of the z matrix.

The entries of the array that are less or equal to one value of z are shown in fig.1 for the inside of the indentation; those for the outside are shown in fig. 2.

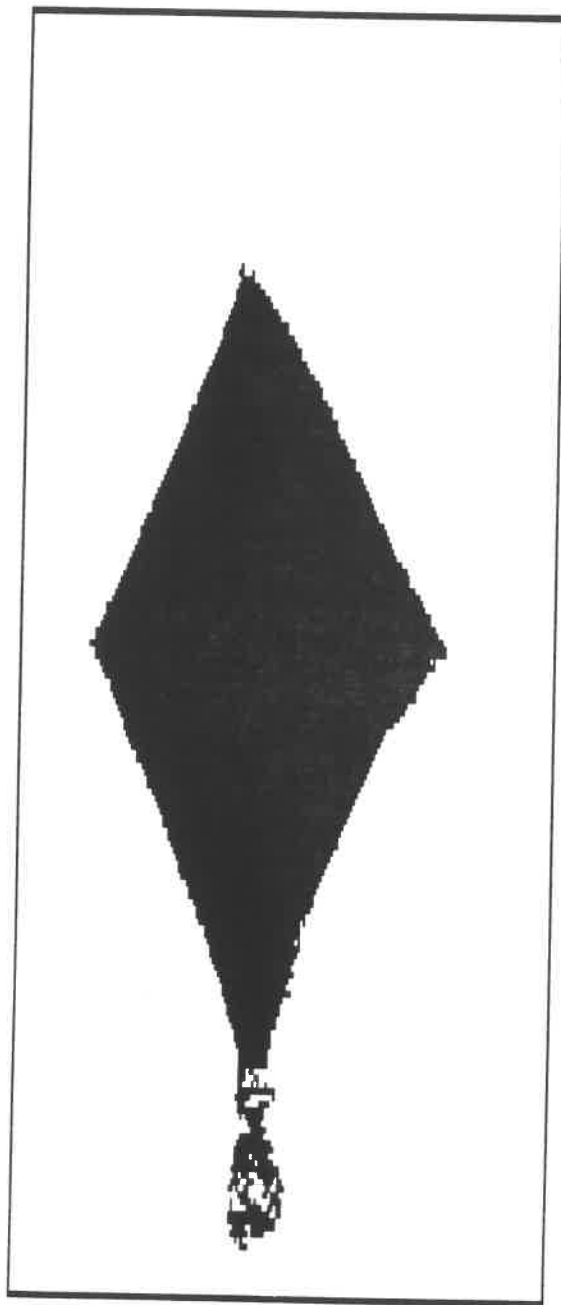


Fig. 1

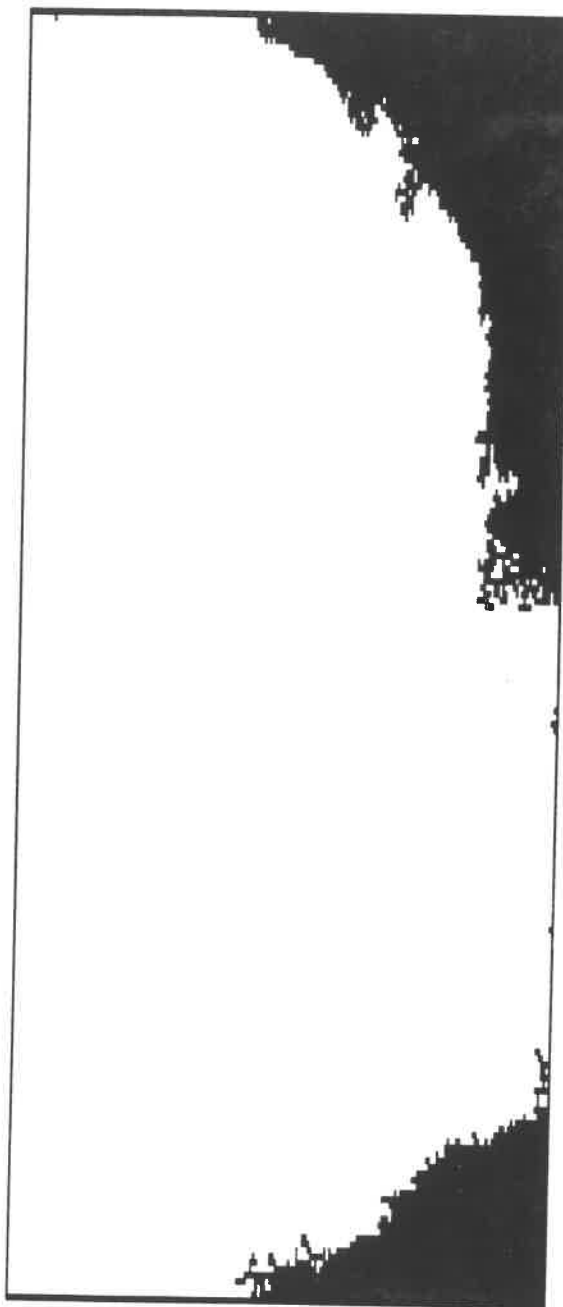


Fig. 2

In order to define the indentation boundary the following parameters were calculated for each z level, the data being grouped in classes:

V/S is the ratio of the volume to the area of the indentation;

$\Delta V = (V)_z - (V)_{z-1}$ is the increment of volume;

$L/2$ is half the side of the indentation (supposed to be a square);

α is the average angle between the vertical axis and the apothem of the indentation;
 F are the frequencies of data.

The graphs of all the above parameters show a peculiarity for the same z value; V/S in particular has a well-defined maximum. We have taken this z value (for which we shall use the symbol ζ) to be the level of the indentation boundary. Fig. 3 shows all these parameters for the indentation represented in fig. 1. The z scale is $z_S = 3000/16385$ and the units are nm. The vertical axis refers to the angle α and the units are degrees. For the other functions the scales were conveniently chosen to display all the graphs on the same figure. For this indentation we found $\zeta = 5222$ by z_S nm.

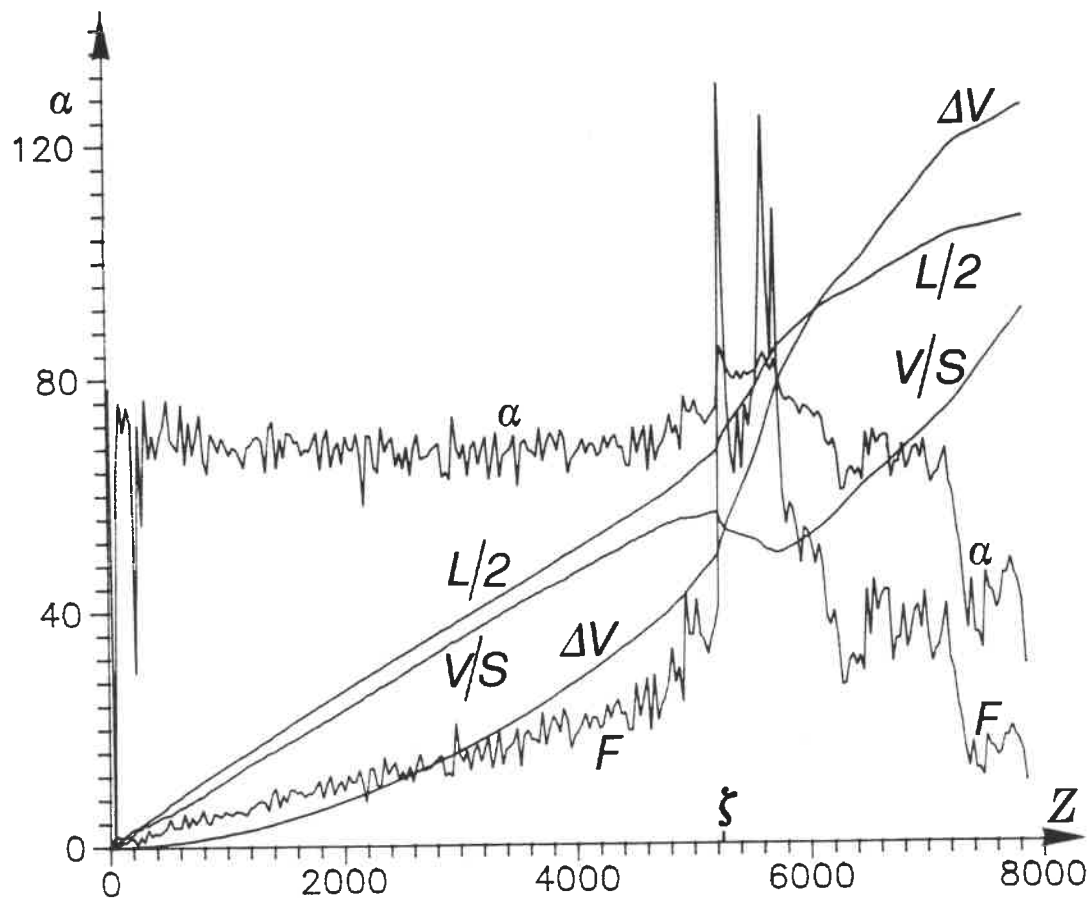


Fig. 3

Fig. 4 shows the contour map of the same indentation and its ζ level. The size of the 200X200 array side is 11000 nm.

Work is in progress on a set of experimental data laid out according to a fac-

torial design to evaluate the contribution (single and combined) of several factors on the uncertainty of the measurement. The following parameters are used: scanning direction, specimen orientation when being indented, specimen orientation when being scanned , X-Y scan size and probe speed.

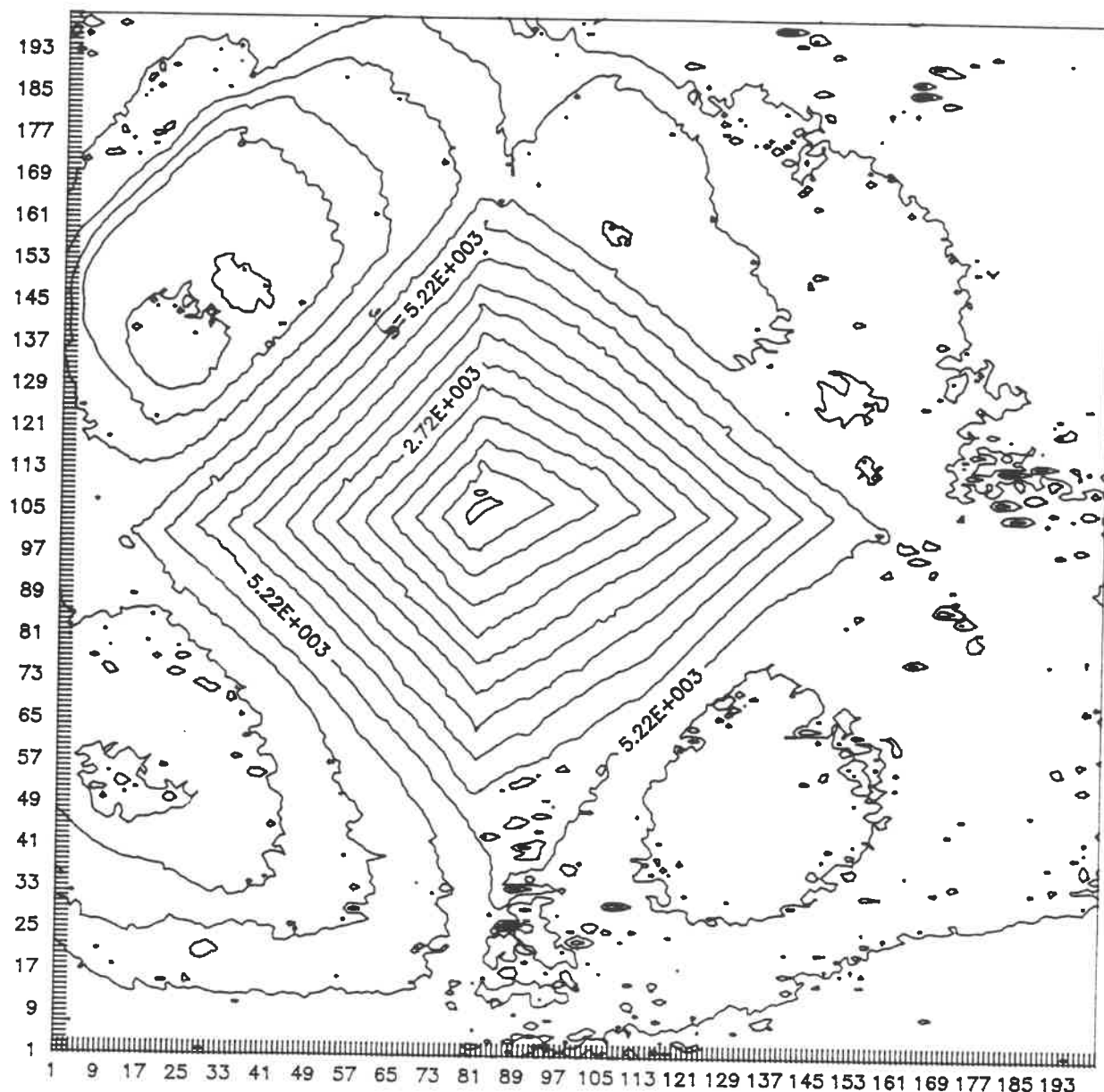


Fig. 4

Aknowledgements

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The determination of Squeeze film damping in capacitance based cantilever force probes

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1 Introduction

A schematical diagram of the force probe sensor analysed in this paper which is a capacitance based sensor of the type reported by Howard and Smith^{1,2} is shown in Figure 1. The probe system consists of a thin cantilever beam that is rigidly attached to a solid base at one end and with a stylus tip attached at the other. The base extends below the cantilever on the face opposite to the tip and is separated by a small air gap. An aluminium film deposited onto the adjacent faces between cantilever and the base form capacitance electrodes.

Theoretically, an ideal probe would be as small as possible with minimal electrode separation. However, as the separation reduces there arises significant squeeze film forces that result in an inherent viscous type damping between the beam and the base.

To fulfil high speed measurement with good fidelity, much attention should be paid to the dynamic properties of the probe. A high natural resonant frequency provides high dynamic response which enables the tip to follow the sharp changes of contours, while damping helps to reduce susceptibility to vibration and may increase general measurement fidelity. However, too much damping makes the dynamic response unacceptably slow, with a corresponding increase in tip/surface forces. Recent investigations of the effect of damping on the fidelity of stylus measurement indicates that optimal dynamic conditions correspond to a system critical damping ratio between 0.4 and 0.7^{3,4}. The ability to calculate specific damping values for a given probe geometry is therefore essential if optimal designs are to be achieved. To evaluate the dynamic characteristics of this kind of force probe, both theoretical analysis and experimental verification of squeeze film forces between the beam and base are presented in this paper.

2 Theoretical analysis

The cantilever on the force probe is modelled as a simple prismatic beam with concentrated mass (diamond tip) on the free end and distributed mass (beam mass) along the beam. Using Rayleigh's approximate method, the fundamental undamped resonant frequency of a cantilever with both distributed and concentrated mass is given by

$$\omega_n = \sqrt{\frac{k}{m_e}} \quad (1)$$

where ω_n is the angular frequency, k is the static bending stiffness of the beam, m_e is the effective mass given by ^{5,6}

$$m_e = m_c + 0.24m_d \quad (2)$$

where m_c and m_d are the concentrated mass (tip mass) and distributed mass (cantilever mass), respectively. Then the probe can be simply modeled as a linear second-order system

$$m_e \ddot{z} + c\dot{z} + kz = F \quad (3)$$

where c is the damping factor dominated primarily by squeeze film action between the cantilever electrodes and F is the contact force applied to the tip.

As previously mentioned, the gap between the electrodes on the cantilever and base block surfaces are made small for high sensitivity of the capacitance sensor. The high aspect ratio between electrode separation and beam dimension makes air in the gap behave as a squeeze film and therefore introduces significant squeeze film forces when beam vibrates. In this case, the squeeze film can be approximated as a low Reynolds number, incompressible and adiabatic film. Then, the reduced Reynolds equation can be used,

$$\frac{\partial^2 p(x, y)}{\partial x^2} + \frac{\partial^2 p(x, y)}{\partial y^2} = \frac{12\mu}{h^3} V(x, y) \quad (4)$$

where p is the gauge pressure distribution in the gap, μ is the viscosity (for air, $\mu = 1.862 \times 10^{-5} \text{ Nsm}^{-2}$), h is the stationary air film thickness, V is the instantaneous velocity of relative motion in the Z direction. The distribution of damping pressure in the air gap is obtained from a solution to Equation (4). Because the free vibration mode shape of the cantilever is independent of time (although its magnitude is changing), the velocity $V(y, t)$ can be represented by two separable functions

$$V(t, y) = V_0(t)f(y) \quad (5)$$

where $V_0(t)$ is the amplitude of instantaneous velocity at the free end and $f(y)$ is given by the deflection mode shape of the cantilever. By bringing Equation (5) into Equation (4), a solving for the appropriate boundary conditions yields the pressure distribution shown in Figure 2. The total damping force acting on the beam can be calculated by

$$F_d = \int \int p(x, y) dx dy$$

$$-\frac{b}{2} \leq x \leq \frac{b}{2}, \quad 0 \leq y \leq l \quad (6)$$

For modelling purposes it is necessary to translate the distributed damping force F_d to concentrated damping force F_e at the same point as the effective mass or probe tip. Using the simple second order model of Equation (3), the damping factor, c , can be derived from the equation

$$F_e = cV_0 \quad (7)$$

where V_0 is the velocity at the free end of the beam (equivalent to $\dot{z}$ in Equation (3)). Using the principle of virtual work,

$$F_e \Delta_0 = \int \int p(x, y) \Delta(y) dx dy \quad (8)$$

where Δ_0 is an assumed small displacement at the end of the beam, $\Delta(y)$ is deflection profile due to Δ_0 , given by

$$\Delta(y) = \Delta_0 f(y) \quad (9)$$

Substituting Equation (7) and (9) into Equation (8), one can obtain the solution for damping factor

$$c = \frac{1}{V_0} \int \int p(x, y) f(y) dx dy \quad (10)$$

$$-\frac{b}{2} \leq x \leq \frac{b}{2}, \quad 0 \leq y \leq l$$

Then the critical damping ratio ξ and damped natural frequency ω_d are calculated by

$$\xi = \frac{c}{2\sqrt{km_e}}$$

$$\omega_d = \omega_n \sqrt{1 - \xi^2} \quad (11)$$

3 Experimental verification and discussion

More than 30 force probes were tested for comparison with predicted results. Some results are listed in Table 1. Comparing these results, a remarkably small difference between theoretical and experimental values can be seen. Modelling and experimental errors may be responsible for these discrepancies.

Modelling errors come from the simplification of the deflection mode. Two deflection models have been used in the calculations, one is static deflection mode with concentrated mass, and the other is the dynamic distortion shape of a simple prismatic beam. No significant difference can be seen from the results (Table 1). In the geometric range of our calculations, the maximum difference between the damping ratios calculated with these two models is 5.58 %. So it can be considered that the damping factors are not very sensitive to the deflection models and, generally, the longer the beam, the smaller the error.

The measured damping ratios and damped resonant frequency are respectively higher and lower than the calculated values for most of probes in experiments. This corresponds to the characteristic of dynamic system, i.e. damping reduces the resonant frequency. In experiments, damping ratios can be affected by many factors, not only air gap and beam dimensions but also viscosity of medium, friction and the rigidity of the mounting between cantilever and base block at the fixed end. All these may influence actual damping ratios. The accuracy of the measurements of geometric parameters will also influence the experiment results and may constitute the most significant error source. The sensitivity of damping ratio to the measurement errors of geometric dimension has been analysed. In the calculation example, 5 % error in the air gap will produce 16.6 % error in damping ratio with 5 % errors in length and width of the beam resulting in the damping ratio errors of 10.6 % and 11.1 % respectively. This indicates that the stringent manufacturing tolerance is essential if damping ratio of the force probe is to be controlled in an expected range.

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Probe	$t(\mu m)$	l (mm)	$h(\mu m)$	ξ_c^*	ξ_c^{**}	ξ_t	$f_{dc}(Hz)$	$f_{dt}(Hz)$	error %
No. 15	195	7.0	10.2	0.925	0.943	-	800	-	-
No. 20	105	6.0	18.6	0.279	0.289	0.325	1132	1130	0.2
No. 23	195	7.5	17.7	0.211	0.215	0.225	1835	1765	4.0
No. 102	105	5.2	14.0	0.445	0.463	0.466	1327	1200	10.6
No. 108	105	6.2	26.5	0.105	0.109	0.148	1111	1100	1.0

ξ_c^* , ξ_c^{**} — damping ratios calculated with static and dynamic deflection modes

ξ_t — measured values of damping ratio;

f_{dc}, f_{dt} — calculated and measured values of damped resonant frequency;

Glass cantilever $E = 70 GPa$, $b = 1.5$ mm, $m_c = 2$ mg; $error = |1 - \frac{f_{dc}}{f_{dt}}| \times 100$

Table 1: The comparison of the results of experiment and calculation

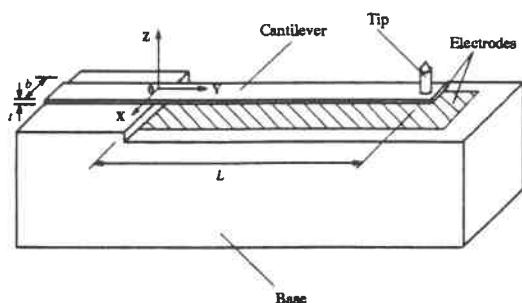


Figure 1: The diagram of the force probe

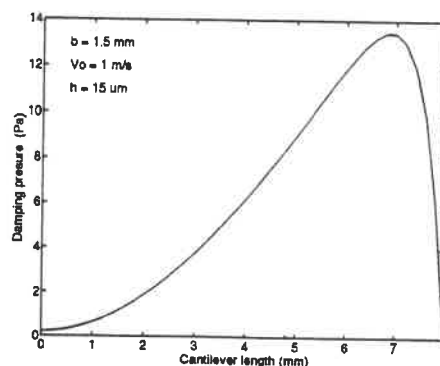


Figure 2: The distribution of the damping pressure along central axis of a cantilever beam, $b = 1.5 mm$, $V_0 = 1 mm S^{-1}$, $h = 15 \mu m$

OPTIMIZATION OF THE LINEARITY AND ACCURACY OF RST MEASUREMENTS

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ABSTRACT

RST technology is a form of vertical-scanning white light interferometric surface profiling that provides a greatly increased dynamic range for the measurement of surface heights compared to conventional phase-shifting interferometry. Measurement resolutions of a few nanometers rms can be obtained over a total measurement range of more than 100 μm peak-to-valley. Maintaining measurement linearity and accuracy over the entire vertical scan becomes difficult with the increasing scan lengths of white light techniques.

TECHNIQUE

The RST measurement procedure is accomplished by translating the vertical axis of the interferometric microscope so that the interference fringes traverse the total peak-to-valley surface height. During the translation, high-speed digital signal processing techniques are used to demodulate the fringe profiles to precisely determine the vertical position of each data point on the surface.¹ Piezoelectric (PZT) devices are used to perform this vertical translation. If the motion of the PZT translator is nonlinear, there will be a corresponding linearity error in the calculated surface height that will affect the accuracy of the measurement.

A solution to this problem can be obtained by controlling the motion of the PZT translator via a linear variable differential transformer (LVDT) position encoder in a servo-feedback configuration. The gain and bandwidth of the feedback control circuit must be optimized to produce the required linear motion. Too high of a gain or bandwidth imposes excessive noise in the motion of the PZT translator (resulting in a loss of repeatability in the measurement), or leads to instability in the control loop. Too low of a gain results in linearity errors that affect the accuracy of the measurement.

Figure 1 shows a diagram of the circuit used for control of the translation, with a closed-loop transfer function whose general form is given by

$$\frac{C(s)}{R(s)} = \frac{G(s)}{1 + G(s)H(s)},$$

where $R(s)$ and $C(s)$ represent the transfer functions of the output signal and reference input respectively, $G(s)$ is the open-loop transfer function, and $H(s)$ is the transfer function of the feedback path.

The system has a finite DC gain in the control loop, represented by $G(s)H(s)$, contains no integration elements, and therefore produces a constant steady-state error in response to a ramp function input during a measurement scan.² This is totally acceptable, since the objective is to produce a constant scan velocity, not an absolute displacement. Low-noise electronics and proper shielding of the cabling connecting the LVDT and PZT are critical to producing good results.

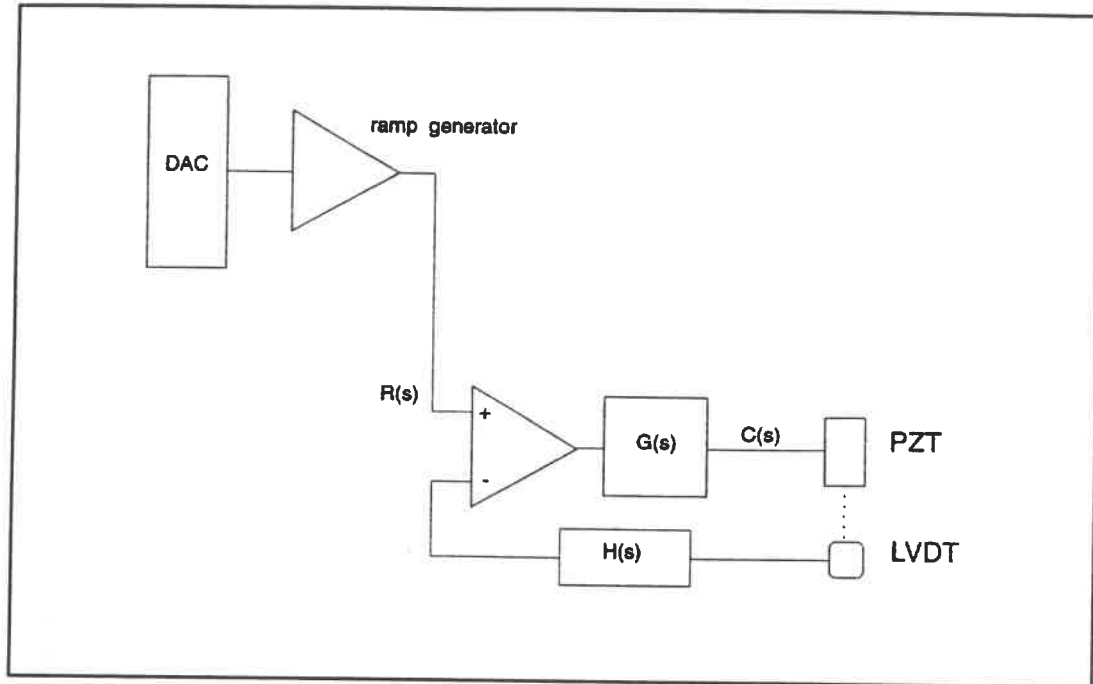


Figure 1. RST PZT control circuitry.

EXPERIMENTAL RESULTS

The linearity of the slope of the LVDT output signal was monitored while scanning the PZT at various speeds and closed-loop gain settings. The goal was to determine the optimum gain/bandwidth combination that would minimize the linearity errors while at the same time introducing negligible loss of repeatability of the measurement due to noise in the feedback loop. The results, plotted in Figure 2, show a sharp improvement in linearity with increasing gain settings. An optimum gain setting occurred at a feedback resistance of approximately 400 k Ω . Beyond that value, no improvement was noted in the linearity error. The bandwidth was then reduced as much as possible while still maintaining linearity over the full range of scan rates required by the measurement.

To verify the measurement accuracy, measurements were made of NIST-traceable step height standards at various closed-loop gain settings. A plot of the measurement error over a range of 1 μm to 50 μm with no feedback compensation is shown in Figure 3. The total measurement error is over 30%. Figure 4 shows a plot of the same measurements done at an optimized closed-loop gain setting. The measurement error is decreased to less than 1% across the entire range, and is actually well within the certified tolerances of each step height standard.³ Increasing the gain produced negligible improvement in the measurement error, indicating that the error is due mainly to the inaccuracies of the step height standards themselves.

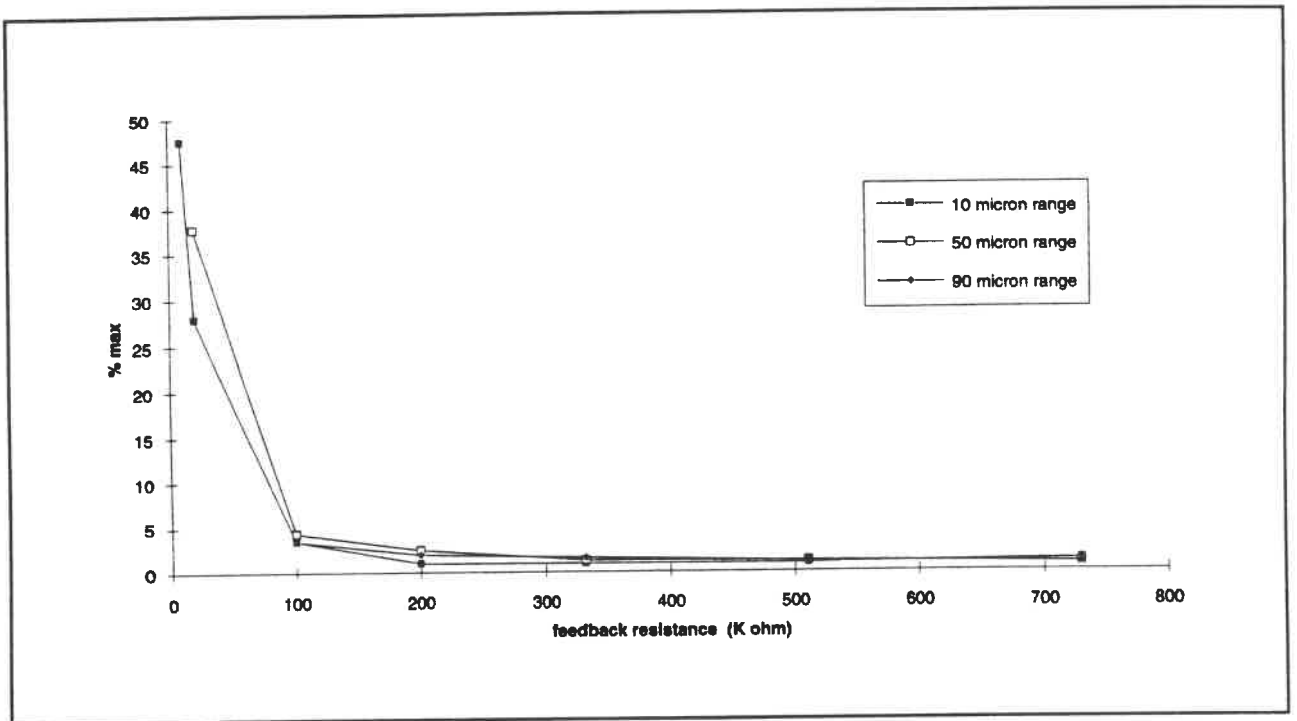


Figure 2. Slope error as a function of control loop gain.

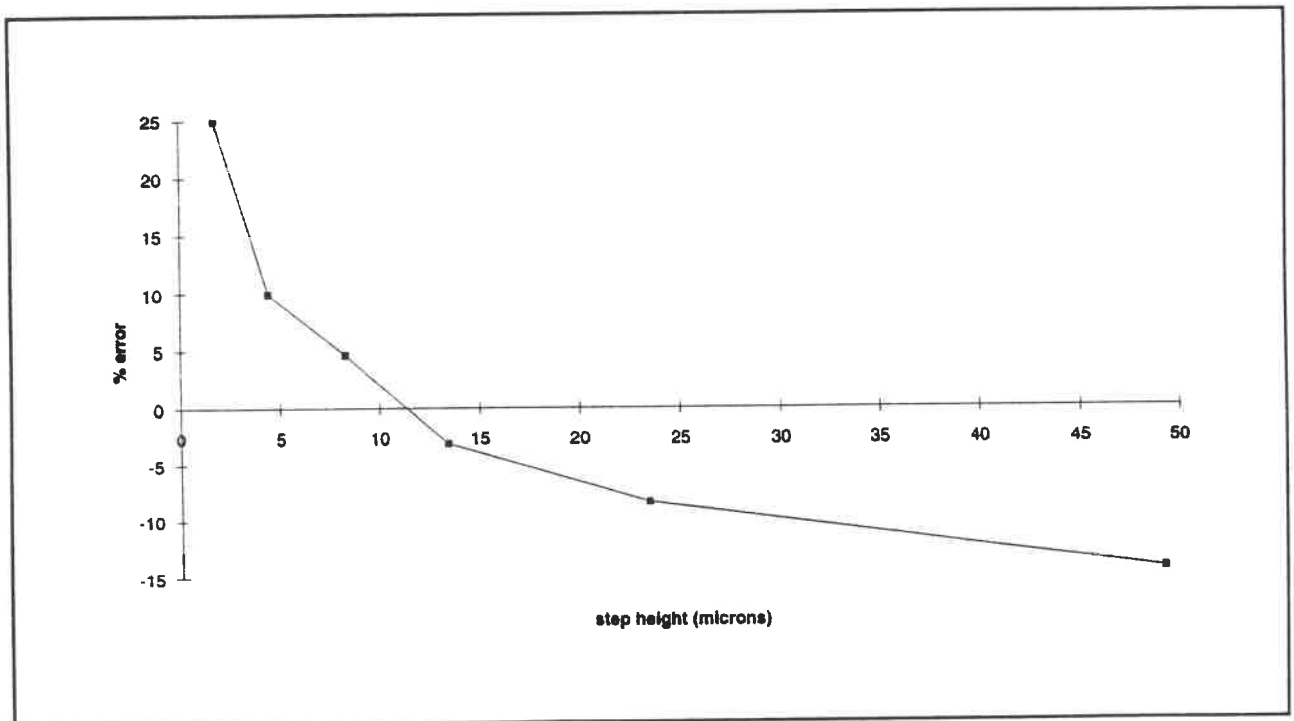


Figure 3. Open loop measurements of NIST-traceable step height standards..

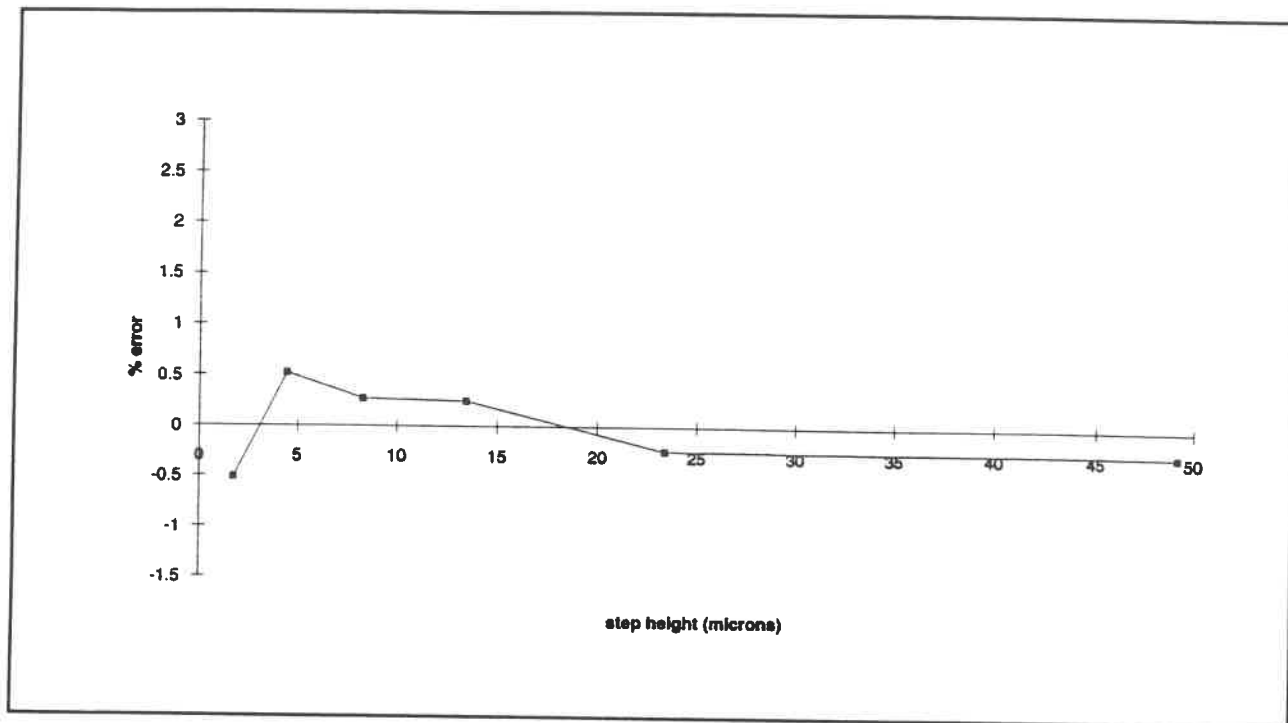


Figure 4. Measurements of NIST-traceable step height standards with optimized closed-loop gain.

Studies were also done to determine the accuracy of the measurement as the initial focus position was varied, to determine effects of gain/bandwidth variations on measurement repeatability, and to evaluate the effect of PZT "rest" or time between successive scans on the measurement accuracy. In all cases, the optimized gain and bandwidth produced the best results. We can therefore conclude that the use of a PZT translator along with an LVDT position sensor in a closed-loop feedback configuration can produce very accurate, repeatable RST measurements provided the gain and bandwidth of the control loop are carefully selected.

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THE FLATNESS MEASUREMENT OF A MEDIUM AND LARGE SCALE MACHINED SURFACE BY USING A LASER BEAM REFERENCE PLANE

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1 Introduction

This paper describes a practical method to measure the flatness of a medium and large scale machined surface by using an optical reference plane which is made by sweeping a collimated laser beam with a precision pentaprism.

The conventional method for flatness measurement of a medium and large scale machined surface is "Union Jack method", in which the flatness of a surface is estimated from straightness measured along some datum lines on the surface. The straightness is measured by using conventional tools such as an autocollimator, a laser interferometer and a theodolite according to the size to be measured. When a machined surface to be measured becomes larger, the procedure of "Union Jack method" becomes very complicated and the uncertainty of measurement increases.

Previously, a practical flatness measurement method has been presented, in which the reference plane is established by sweeping a collimated laser beam in a plane parallel to the surface to be measured⁽¹⁾. In this paper, the measurement accuracy of the method is theoretically analyzed and, based on the analysis, prototype equipments are made for the flatness measurement of medium and large scale machined surfaces.

2 Analysis of the Geometrical Error of Reference Plane Made by Sweeping a Laser Beam

As shown in Fig.1, the orthogonal coordinate system o-xz is taken in the principal plane of a pentaprism. The incident laser beam I_1 is set approximately in parallel with the z-axis and when the pentaprism is rotated precisely around the z-axis, the reflected beam I_2 sweeps in the plane normal to the incident beam I_1 . The plane formed by beam I_2 can be used as a reference plane in flatness measurement. At this time, what exert influence on the accuracy of the reference plane are the following three factors:

- (a) error in the apex angle θ of the pentaprism;
- (b) angular deviation of I_1 from z-axis;
- (c) error motion in the rotation of a pentaprism.

In Fig.1, the reflection planes of the pentaprism are denoted M_1 and M_2 and the direction cosines of their normals $(\lambda_1, \mu_1, \nu_1)$ and $(\lambda_2, \mu_2, \nu_2)$, respectively. Also, the direction cosines of I_1 are denoted by $(\alpha_1, \beta_1, \gamma_1)$ and those of I_2 by $(\alpha_2, \beta_2, \gamma_2)$. Then, the following geometric relations are derived:

$$\left. \begin{aligned} \alpha_2 &= \alpha_1 \cos 2\theta + \gamma_1 \sin 2\theta \\ \beta_2 &= \beta_1 \\ \gamma_2 &= -\alpha_1 \sin 2\theta + \gamma_1 \cos 2\theta \end{aligned} \right\} (1)$$

Generally, I_1 and I_2 do not intersect each other, but the distance between the two beams is very small. Hence, the angle ψ of intersection between I_1 and I_2 can be

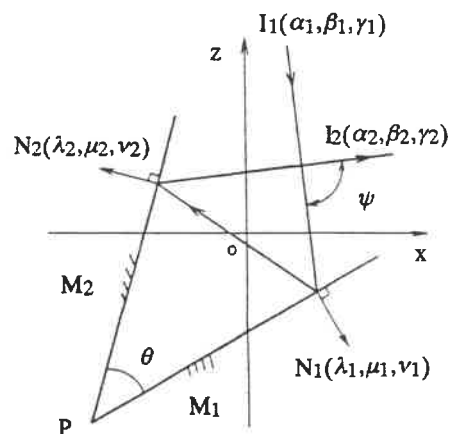


Fig.1 Ray trace in pentaprism

represented by:

$$\cos \psi = \alpha_1 \alpha_2 + \beta_1 \beta_2 + \gamma_1 \gamma_2 . \quad (2)$$

Then, from the equations (1) and (2)

$$\cos \psi = \beta_1^2 + (1 - \beta_1^2) \cos 2\theta \quad (3)$$

is obtained. At the ideal case of $\beta_1 = 0$ and $\theta = \pi/4$, the intersecting angle between I_1 and I_2 is given by $\psi = \pi/4$, which is the well known feature of a pentaprism. When a error $\Delta\theta$ exists in the apex angle of a pentaprism and the intersection angle by $\Delta\psi_1$ accordingly, that is:

$$\left. \begin{aligned} \theta &= \frac{\pi}{4} + \Delta\theta \\ \psi &= \frac{\pi}{4} + \Delta\psi_1 \end{aligned} \right\} \quad (4)$$

then

$$\Delta\psi_1 \approx \beta_1^2 - 2\Delta\theta . \quad (5)$$

Equation (5) indicates that, if $\Delta\theta > 0$, the intersection angle can be made as $\psi = \pi/2$ by adjusting only β_1 so that $\beta_1^2 = 2\Delta\theta$.

The error motion of the rotating pentaprism is thought to be divided into following two items:

- (i) angular deviation of the rotation axis from the z-axis;
- (ii) radial and axial deviations of the pentaprism.

With regard to (i), its influence on ψ is clearly the same as that in the case when I_1 is not in parallel with the z-axis. Therefore, when the angle made between the projection of the rotation axis on the yz-plane and z-axis is β_0 , its effect $\Delta\psi_0$ on ψ is obtained from equation (5) as:

$$\Delta\psi_0 \approx \beta_0^2 . \quad (6)$$

Radial and axial deviations, e_r and e_a respectively, have no influence on ψ , but they change the position of I_2 as shown in Fig.2. The total deviation Δz of I_2 along z-axis due to e_r and e_a is given as:

$$\Delta z \approx e_a - e_r \quad (7)$$

When the position of I_2 is to be detected at the point A that is apart from the pentaprism by l , as shown in Fig.3, the overall error of detection Δh is represented by:

$$\Delta h \approx \Delta\psi l + \Delta z \quad (8)$$

where

$$\begin{aligned} \Delta\psi &\approx \Delta\psi_1 + \Delta\psi_0 \\ &\approx \beta_1^2 - 2\Delta\theta + \beta_0^2 . \end{aligned} \quad (9)$$

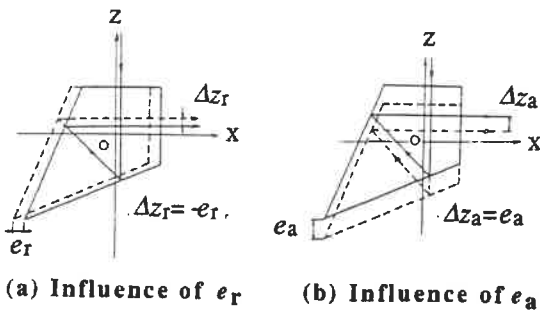


Fig.2 Influences of radial and axial deviation on the position of reference plane

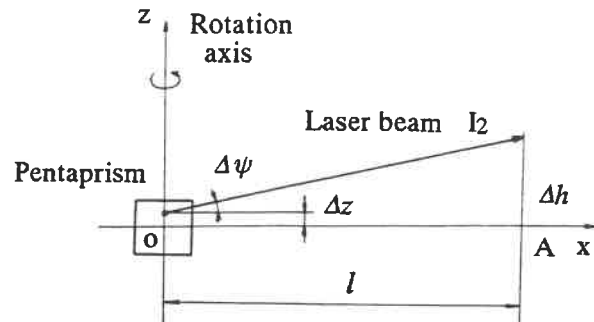


Fig.3 Total error of reference plane

From these analysis, it is known that strict specifications should be applied to $\Delta\theta$, e_r and e_a because β_1^2 and β_0^2 are negligibly small.

3 Flatness Measurement

As the first application of the presented method, the machined surfaces of the main member of a tower for a large suspension bridge have been measured. Figure 4 shows the outline of the tower and its main member. The member is a box-shaped welded steel structure of about 7m X 11m X 9m and the weight exceeds 200 tons. Thus, rectifying work in a construction site is extremely difficult, accordingly very strict dimensional and geometrical accuracy control is required in manufacturing the members in a workshop. In particular, the flatness of the surfaces at both ends and their parallelism are important whose specified values are 0.2 mm and 20 arcsec, respectively.

The measurement system is schematically shown in Fig.5. The laser beam A is set in parallel to z-axis of the machine tool. Changing the direction of this beam with pentaprism-1 and -2 by a right angle, the beams B and C are obtained. By turning the pentaprisms around the optical axis of the beam A, sweeping of the beams B and C is carried out in the planes parallel to the XY-plane of the machine tool. By detecting the distance between the points on the end surfaces 1 and 2 of the member and the centers of the beams B and C with position sensors⁽²⁾, the flatness of each and the parallelism of both end surfaces are measured. The error $\Delta\theta$ of the pentaprism used in this system is smaller than 1 arcsec and the motion error in rotation are $e_r = 5 \mu\text{m}$, $e_a = 12 \mu\text{m}$ and $\beta_0 = 15$ arcsec. One of the results of flatness measurement is presented in Fig.6. For reference, the values measured with a theodolite are additionally written in the parentheses.

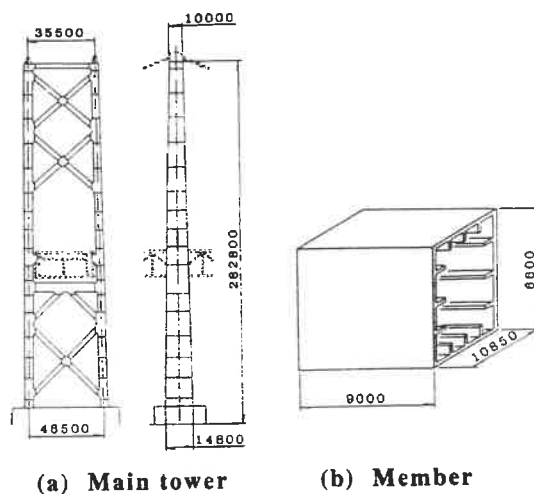


Fig.4 Main tower of large suspension bridge and its member

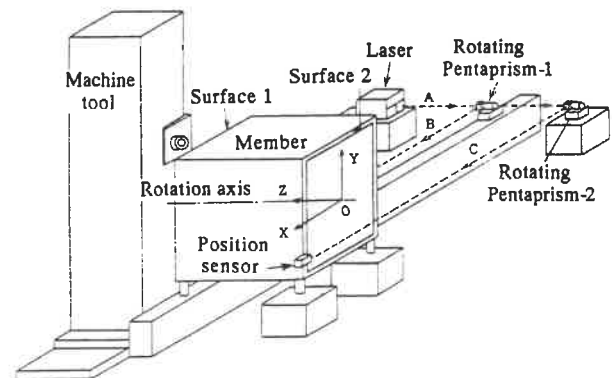
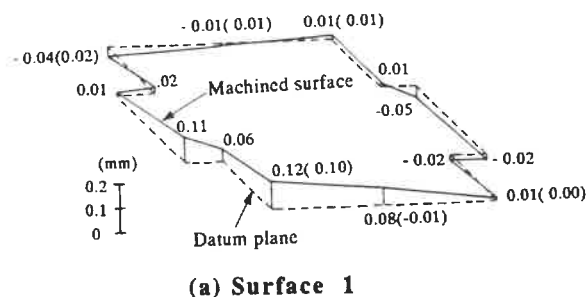
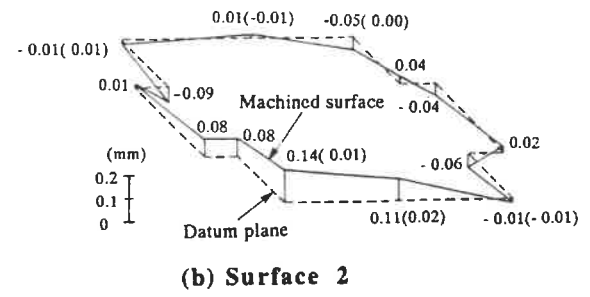


Fig.5 Measurement system for member of large suspension bridge



(a) Surface 1



(b) Surface 2

Fig.6. Measurement results of member of large suspension bridge

Secondly, this method has been applied to measuring the flatness of medium size surface, that is, a precision surface plate of 1 m X 1 m. The measuring system is schematically shown in Fig.7. The pentaprism and the spindle used here are more precise than the previous ones. $\Delta\theta$ of the pentaprism is smaller than 0.5 arcsec, and both of e_r and e_a are smaller than 0.3 μm . β_0 is about 5 arcsec. The total measurement accuracy of the system is about 10 μm which is mainly governed by the detection accuracy of the position sensor used, a CCD camera. The measurement accuracy will be improved to a large extent by refining the position sensing method. The obtained result is shown in Fig.8 comparing with that obtained by the conventional "Union Jack method" using laser interferometer.

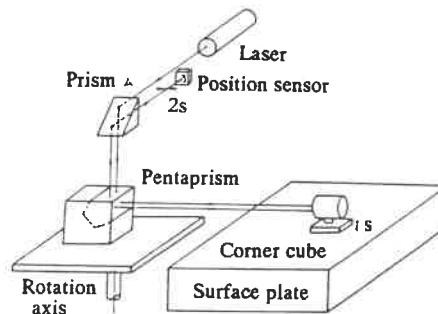


Fig.7 Measurement system for medium surface plate

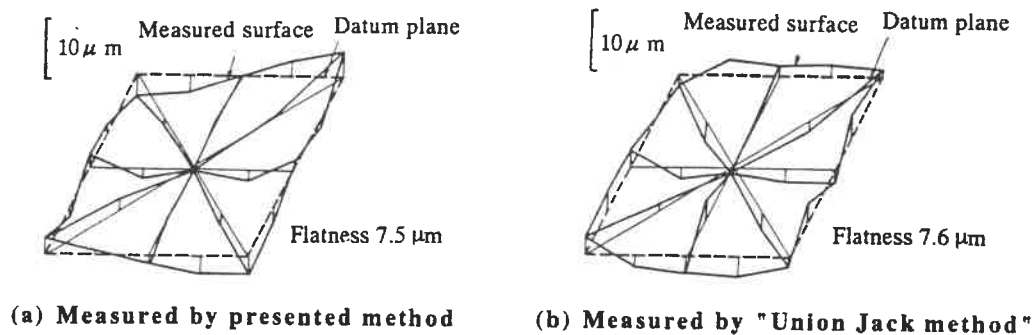


Fig.8 Measurement result of medium surface plate

4 Conclusion

An optical reference for measuring the flatness of medium and large scale machined surfaces has been presented. The reference is made by sweeping a collimated laser beam with a rotating pentaprism. The geometrical accuracy of the reference has been analyzed with respect to the geometrical error of a pentaprism and its rotational error motion, and it is clarified through the theoretical analysis that the proposed optical reference shows its advantage extremely when the size of a surface to be measured is larger than a few meters square. Based on this analysis, two measurement systems have been designed, one is for measuring a very large surface of about 11 m X 7m and the other for a medium size surface of 1m X 1m. The obtained results have verified the accuracy analysis and the feasibility of the proposed optical reference.

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AUTOMATED DIAMETER AND STRAIGHTNESS MEASUREMENTS OF LONG, SMALL BORES

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Introduction

Air gages are commonly used for diameter measurements, and a less common application is for straightness measurements, where a special staggered jet air probe is employed. The benefits of air gaging are high resolution, speed, non-contact, and the ability to reach into long bores. Typically, air gages are used manually, or with simple fixturing, and interpretation of the measurement readings is often left to the judgement of the operator. Measurement of a long part might involve the use of shims or a template to find the next measurement position. By this method taking diameter measurements is tedious but straightforward. Straightness measurements, on the other hand, were difficult, if not impossible to interpret. In our quest to make high-precision air dielectric, coaxial transmission lines (airlines) for the calibration of microwave network analyzers we needed a reliable method of obtaining straightness as a function of length and diameter as a function of length data for process control and performance modeling purposes. This led to the development of a PC-controlled Automated Air Gage Measurement System.

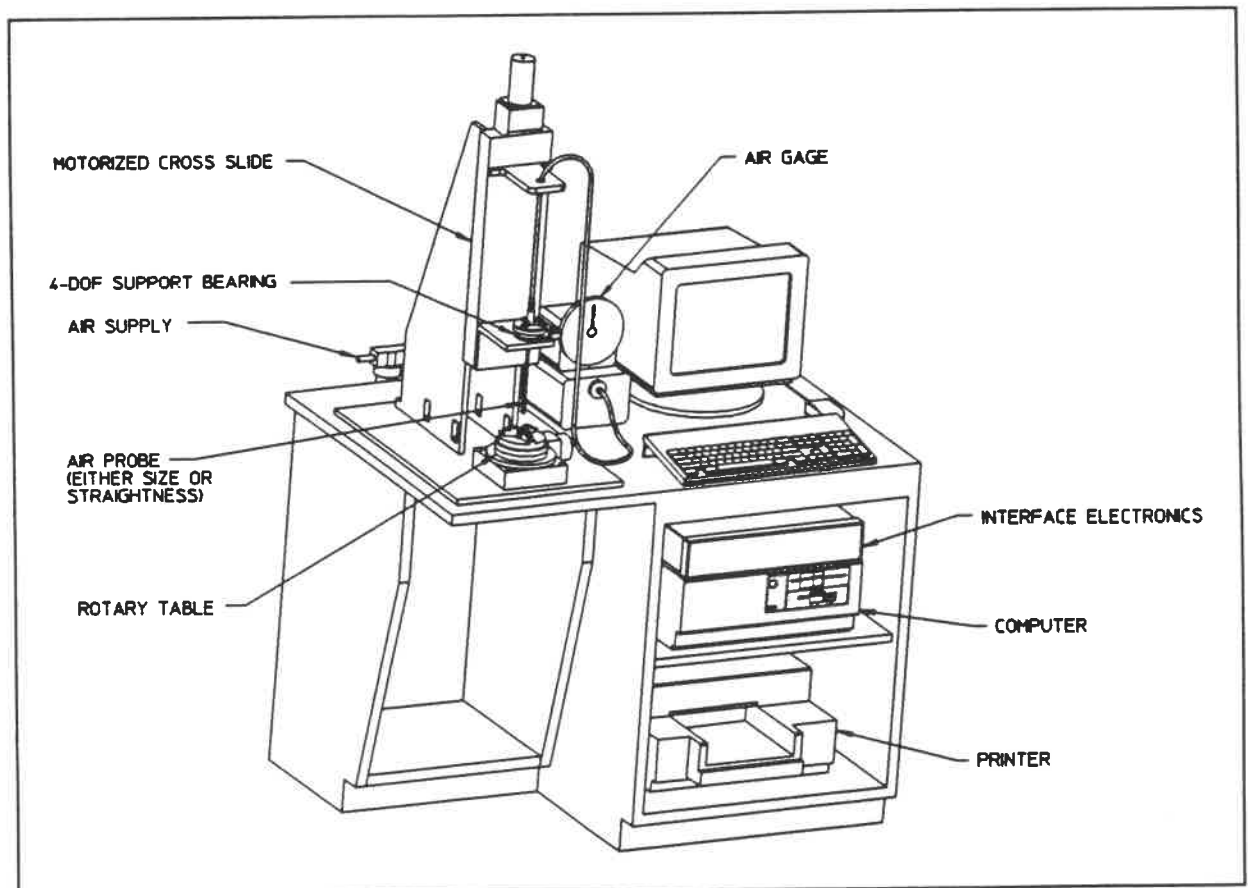


Figure 1

System Description

The Automated Air Gage Measurement System is a full-featured inspection workstation for two-dimensional bore metrology problems. See Figure 1. The system combines an axial/rotational positioning system and a high-precision air gage with signal filtering under flexible, menu-driven computer control, so that difficult bore gaging problems, once restricted to the metrology lab, can be performed routinely in a production environment. The part or part fixture is held in a scroll chuck, and measurements are taken in a vertical orientation to minimize misalignments due to gravity. A linear positioner with a stepper motor drive is used to lower an air probe into the part. A special four degree-of-freedom bearing design is incorporated where the probe handle is clamped to allow the probe to self-center in the bore. A rotary table is used to index the part 360° at each measurement location. What was once a difficult manual task requiring the "proper touch" to get accurate results is now automated and repeatable. Timely and very accurate bore diameter, taper-barrel-bellmouth, and ovality results can be provided to the production area for process control, capability studies, and process improvements. In addition, for the first time, the straightness of long bores can be conveniently and accurately quantified. A graphical output of bore measurement results is displayed on the screen and can be printed for hardcopy. Data are stored on a database and can be read into statistical process control (SPC) software or a spreadsheet for further analysis.

Bore Straightness Measurement

In order to characterize the straightness of small (1.85mm - 7mm), deep (40mm - 300mm), through bores in our products, we use an air straightness gage. This gage has 4 orifices, two on one side and two on the other. The first two are placed one inch apart, and the other two are as close together as possible. See Figure 2.

At any depth in the bore, the air gage can measure the straightness over one inch (incremental straightness) at a given angle relative to some datum established by the product. The maximum incremental straightness at a particular depth in the bore is found by rotating the part relative to the air gage and finding the largest deviation from zero. In practice, the manipulation is done with the Automated Air Gage Measurement System and the maximum point is found by fitting the data to a sine wave, where the amplitude of the sine wave is twice the straightness deviation.

The specifications of our products include the straightness of the bore along its entire length.

This number is the overall straightness. Therefore, the incremental straightness data must somehow be combined so that the overall straightness can be computed.

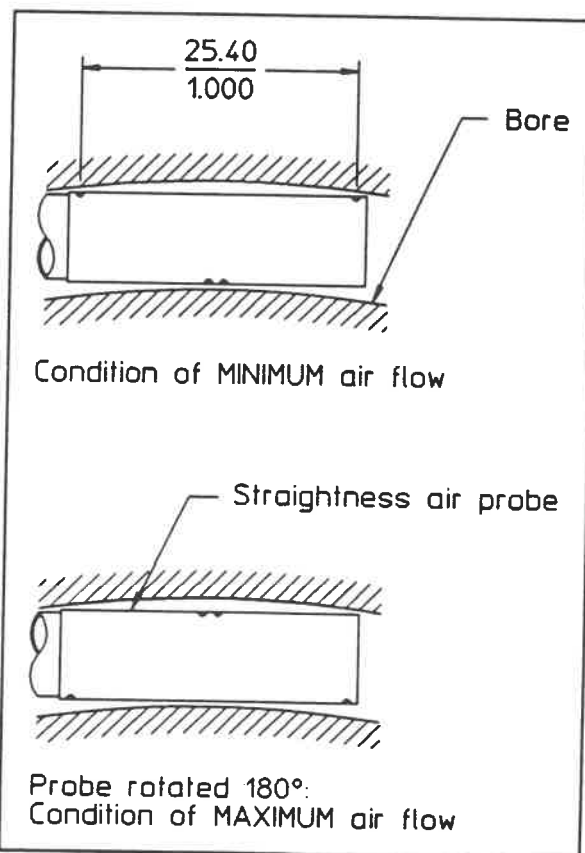


Figure 2

The solution is borrowed from the theory of elasticity, which solves a similar problem in beam theory. Figure 3 shows an expression for curvature, and figure 4 relates the curvature to the incremental straightness measured by the air probe.

The equation at the bottom of Figure 3 is numerically integrated to give the straightness deviation (y) with respect to the length (x) given the incremental straightness (a) and the straightness probe chord length (c).

All of the part and probe manipulation, the calculations, the digitizing of the air gage data, the graphical display of straightness, and the archiving of results are done by the Automated Air Gage Measurement System.

Verification of Straightness Results

A comparison test was performed which compared the measurements made on the Automated Air Gage Measurement System with measurements using a Zeiss UPMC 550 Coordinate

Measurement Machine. Special airlines were fabricated with ground reference outside diameters. These airlines were first measured on the Automated Air Gage Measurement System. The parts were then sectioned using WEDM and then measured on the Zeiss UPMC 550 CMM. The offsets of the center of the inside diameter with respect to the outside diameter were compared to the expected offsets using the Air Gage System. Agreement in the magnitude of the offset was typically below 3 μm , and the worst case was below 5 μm .

Diameter Measurements

In contrast to the straightness measurements, the diameter measurement routine collects and displays data without a great deal of calculations involved. The part is rotated through one revolution at each location, while data is collected through the air gage. A maximum, average, and minimum is recorded at each location and plotted on the screen. The resolution is quite good, and accuracy is limited by the quality of ring gages presently available. See Figure 5.

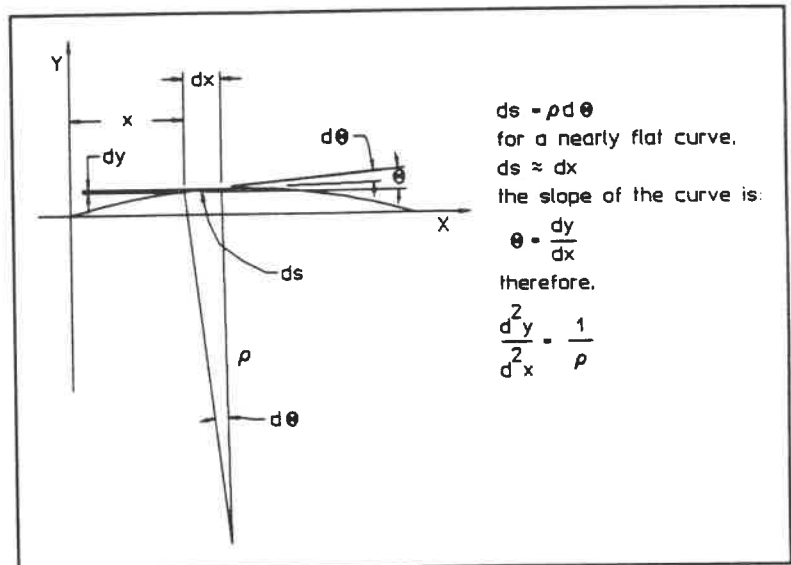


Figure 3

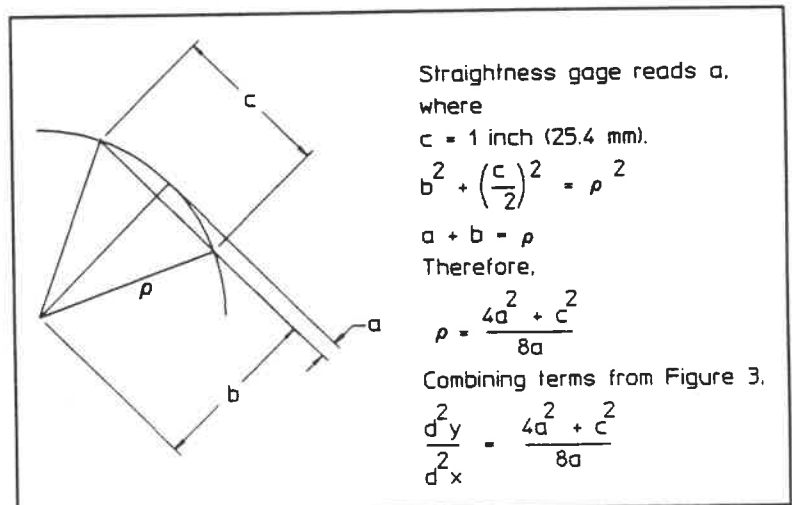


Figure 4

Sample Output of Diameter vs. Length Capability

Measurements were taken on a 7.003 mm Class XX ring gage. The thickness of the gage was 12.7 mm (0.500"). Allowing for air gage bleed off at the entrance and exit of the hole, 11 mm of bore length was probed. Calibration took place 1.0 mm in from the first reading. The measurements revealed form deviations in the ring.

Key:

- - - Max/Min readings at each location
- Mean (Avg.) diameter at each location

Consistency = $\text{Max(Avg.)} - \text{Min(Avg.)}$

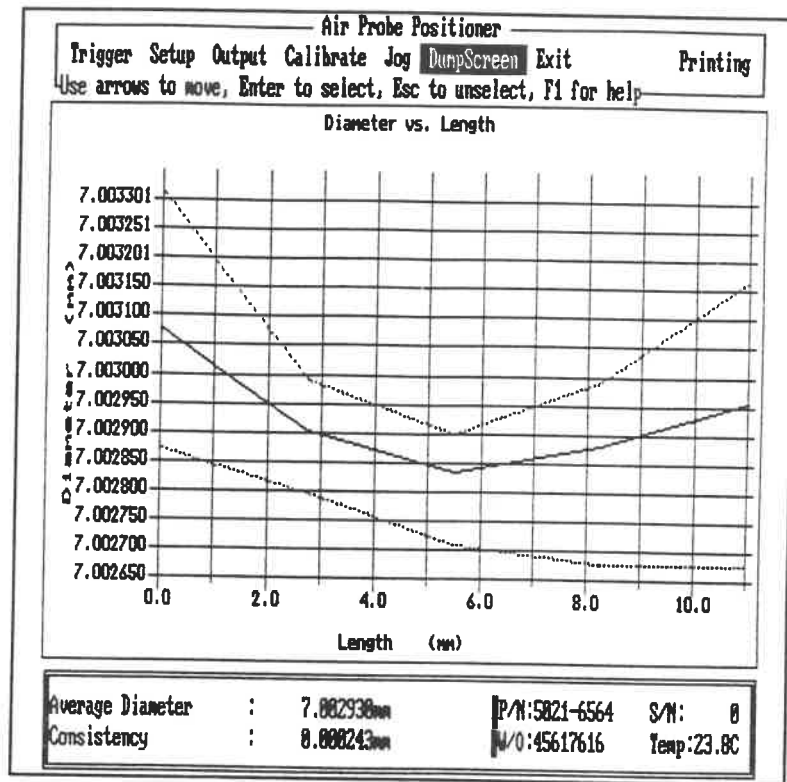


Figure 5

Sample Output of Straightness Capability

Measurements were taken on an actual product during a production run. In this case the straightness of a gundrilled hole was being monitored. The software is capable of piecing together a series of Incremental Straightness measurements from the various axial and rotational positions into a picture of Overall Straightness of the hole and exit position of the drill.

Key:

Max Between Centers predicts maximum runout of the hole for a part held between centers.

Max Incremental shows the worst case incremental straightness.

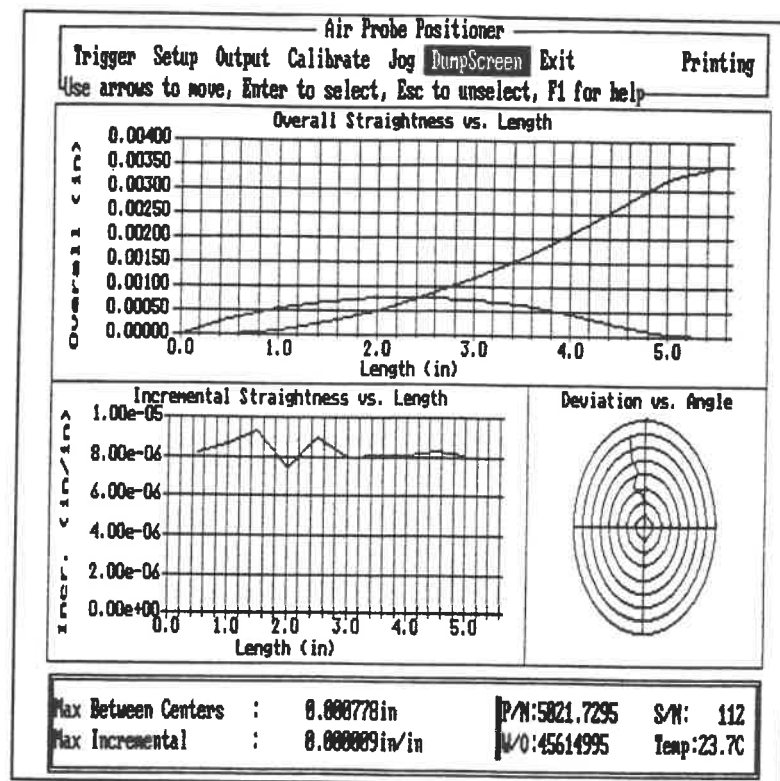


Figure 6

Genetic Search Methods for Assessing Geometric Tolerances

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There are two broad categories of geometric form assessment: measurement of the deviation of a data-set from a best fit reference shape; and inspection of whether a data-set lies totally within a permitted tolerance zone (acceptance testing). Mathematically, both are types of constrained optimization. The latter requires more constraints but need not be driven to optimality: it is enough to show whether a feasible solution exists. Some inspection problems are naturally linearly constrained and some others can be well-linearized, and for these a special algorithm of reasonable efficiency can be applied [Carpinetti and Chetwynd, 1992]. Linearization of other inspection problems may involve quite large degrees of approximation and in these cases the trade-off is unclear between accepting an approximation in the formulation of the constraints or using accurate constraints and much greater computational effort to obtain a potentially more definitive result. Solution methods are required that will perform well with the level of computer power likely to be dedicated to an individual measuring machine.

Here, we explore the application of Genetic Search methods to the determination of the feasibility of the inspection problem. The use of genetic search in non-convex, non-linear constrained optimization problems is reported by Hajela [1990]. Genetic algorithms are different from typical search methods in the sense that they search from a population of points, not a single point, and they use probabilistic transition rules, not deterministic rules [Goldberg, 1989]. An obvious advantage in this approach is that the search is not based on gradient information, and has, therefore, no requirements on the continuity or convexity of the design space. The corresponding disadvantage is that many iterations may be required as special features of the decision surface are not exploited. Since our problem requires only to find a feasible, and not an optimal, solution the advantages may outweigh the disadvantages.

Although software implementations of the genetic algorithm are available, there appear to be no clear-cut ways of checking their efficiency with problems of our type other than by experimentation. Hence a Genetic Search model was set up and suitable values for its control parameters were checked experimentally. A public domain software was used for performing the tests: Genesis (Genetic search implementation system, version 5.0), for function optimization developed by Grefenstette [1990], written in C language. One generation comprises the following procedures: selection, crossover, mutation and evaluation. The user must write an evaluation procedure, which calculates for each solution its fitness value based on the objective function to be optimized. In our case it evaluates whether a solution X is feasible and applies a penalty factor proportional to the degree of infeasibility according to $r \sum_{i=1}^m [g_i(X)]$, where m is the number of violated constraints $g_i(X)$, and r is a penalty term. The algorithm searches through infeasible solutions in an attempt to minimize the penalty term. The iteration process terminates when the penalty term is reduced to zero, meaning that a feasible solution has been found. However, in cases where no feasible point exists the penalty term is never reduced to zero and hence the iteration process has to be stopped. In this case, even though there is no formal proof of infeasibility, the method may still be of practical use since if no feasible point is found within a specified number of generations the possibility of a false negative result by terminating as a failure will be acceptably low.

In order to evaluate the efficiency of this model under practical conditions, two inspection problems were considered. The first was inspection of roundness and centre position of a hole,

formulated as: for a data set $(x_i, y_i), i = 1, \dots, N$, find (a, b) and R such that:

$$\begin{aligned} (x_i - a)^2 + (y_i - b)^2 &\leq (R + t_r)^2 \\ (x_i - a)^2 + (y_i - b)^2 &\geq R^2 \\ (x_o - a)^2 + (y_o - b)^2 &\leq t_{cp} \end{aligned} \quad (1)$$

where t_r and t_{cp} are the roundness and centre position tolerances and (x_o, y_o) the nominal centre position (specified in design). The second was inspection of position and dimension of four holes, of the same dimensions and tolerances, forming a square frame on a plate. This can be formulated as: find reference centres $(a_k, b_k), k = 1, \dots, 4$ such that:

$$R_{min}^2 \leq (x_i - a_k)^2 + (y_i - b_k)^2 \leq R_{max}^2, \quad i \in I_k, \quad k = 1, \dots, 4 \quad (2)$$

and

$$\begin{aligned} (L - t_{cp})^2 &\leq (a_1 - a_2)^2 + (b_1 - b_2)^2 \leq (L + t_{cp})^2 \\ (L - t_{cp})^2 &\leq (a_1 - a_3)^2 + (b_1 - b_3)^2 \leq (L + t_{cp})^2 \\ (L - t_{cp})^2 &\leq (a_2 - a_4)^2 + (b_2 - b_4)^2 \leq (L + t_{cp})^2 \\ (L - t_{cp})^2 &\leq (a_3 - a_4)^2 + (b_3 - b_4)^2 \leq (L + t_{cp})^2 \end{aligned} \quad (3)$$

where R_{min} and R_{max} are the lower and upper radius dimension limits and t_{cp} the centre position tolerance.

A set of three circular profiles, drilled, milled and bored holes, were measured on a coordinate measuring machine in manual mode. A set of 30 points evenly but not exactly spread around the circular features were data-logged and stored in disk files. Additional simulated data sets were generated by a general periodic wave of the form $r(\theta) = A \sin N\theta + r_o$ (where N is the number of lobes, A the amplitude and r_o the nominal radius) [Damir, 1979]. An element of noise was introduced by adding to the signal a random number multiplied by an amplitude parameter. MATLAB [The MathWorks, 1992] was used to generate these data sets. For the single hole problem, three data sets were generated, each one of thirty evenly sampled points around 360° , with three, four and five lobes, the amplitude parameter A set in the region of 1.0×10^{-2} and the amplitude parameter of the noise in the order of 1.0×10^{-3} . For the grid of holes, three sets of four data sets were generated, again with three, four and five lobes, and amplitude parameter A set in the region of 1.0×10^{-1} and the amplitude parameter of the noise in the order of 1.0×10^{-3} . These figures were chosen so as to simulate units in millimeters.

The dimension, location and out of roundness of the holes were first measured by using a minimum zone circle (MZC) best-fit criterion. Tests were then performed for each data set by simulating tolerance values based on the measured errors, and feeding these tolerance values and the data points to the genetic algorithm, so as to measure the number of generations and the computation time to get a feasible solution, if one existed. The effectiveness of this method was then compared with that of formal non-linear programming techniques by using a off-the-shelf software implementation of the Generalized Reduced Gradient method (GRG) [GINO, 1992] and performing the same set of tests.

Preliminary tests were performed in order to set control parameters of the genetic search model such as population size, lengths of strings representing each variable, crossover and mutation rates and initial population. These tests indicated that the algorithm performs best by setting the population size between 60 and 80, the string length as large as it can be and crossover and mutation rates in the region of 0.65 to 0.75 and 0.001 to 0.01 respectively. The iteration process is reduced by defining the initial population randomly from a range of values as narrow as possible around the expected solution. For the problems in consideration, a good estimate of the solution is given by the least squares method. The difficulty with overly narrowing the range of candidate

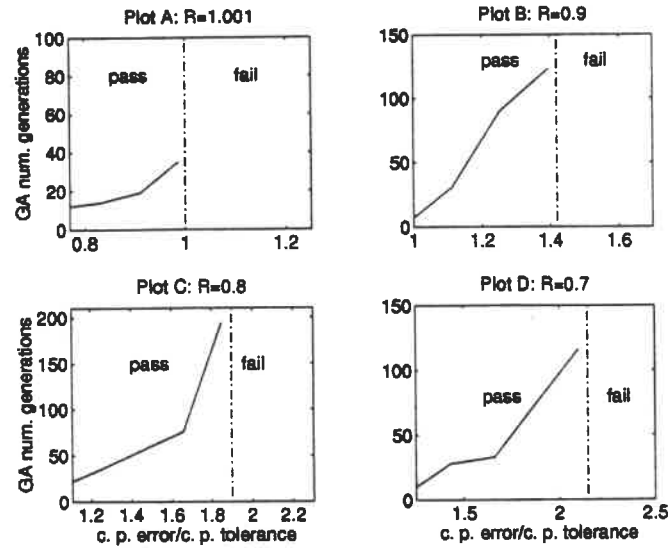


Figure 1: Number of generations to get a feasible solution for different ratios of centre position error to tolerance, and different ratios, r , of the difference between the MMC radius defined by the MZC reference and the specified nominal radius to the tolerance on radius. Buffer zone of 1.0×10^{-5} mm.

solutions is that the only acceptable solutions may lie outside the chosen range. In order to avoid this, the proposed strategy is: find the least squares solution for each circular feature; for each variable define a range of candidate solutions of width equal to the tolerance zone, such that its least squares value lies midway between the lower and upper limits.

The number of iterations of the genetic search model so established was plotted as shown in figure 1 for a typical data set of the second problem. Plots A, B, C and D correspond to different ratios r of the difference between the MMC (Maximum Material Condition) radius defined by the MZC reference and the specified nominal radius to the tolerance on radius. The vertical dashed lines are an approximated indication of the point at which the problem becomes infeasible. It can be seen from graphs B to D that when the actual hole diameter is larger than its MMC size, the combination of features can be acceptable in terms of interchangeability even though the eccentricity of some of the features (when measured using a best-fit criterion) is larger than its corresponding tolerance. This is a clear advantage of this approach, when compared with the standard method of assessing the parameter values of best-fit geometric elements. In general, the number of generations for cases that easily pass inspection are around 50 for the second model problem (as shown in figure 1) and well below 50 for the single circular feature problem. However, when the error becomes closer to the tolerance value, that is, as the feasible region shrinks, the number of generations tends to increase exponentially. In the limiting case, when the error equals the tolerance value and the feasible region reduces to a point in space, the number of generations to find this point is unaffordably high. This can be overcome by using a buffer zone on the boundaries of the feasible region, such that the feasible region is effectively enlarged by a certain amount. The results shown in figure 1 are for a buffer zone of 1.0×10^{-5} (for units given in millimeters). The number of generations in the limiting cases is then usually 200 to 300 for the second model problem. This represents a computation time of no more than few seconds on a computer of moderate speed. In practice, introducing a buffer zone implies a slight widening of the geometric tolerance zone. Typically the increase might be one part in a thousand, so the error introduced by this method will be no more

than 0.1 %. Provided the buffer is small compared to the tolerance band the likelihood of falsely accepting bad components will be tolerably low. Of course, the buffer zone could be taken off the tolerance, with a slight possibility then of false negative results in marginal cases. Therefore, by keeping the tolerance zone well below the geometric tolerance value, this is likely to represent no serious problem.

When there is no feasible solution, the search would continue for ever and it has to be stopped artificially. However, it is possible to predict the cases of easy rejection by monitoring the rate at which the objective or fitness function of the problem changes, that is by monitoring the evolution towards feasibility of the best candidate solutions from generation to generation. A strategy for inspecting using this method might involve the definition, during a setup phase, of two threshold points, as follows: the first threshold point is defined by monitoring the evolution of the best solutions towards feasibility such that if the fitness function for the best solution has not reached a certain value after a certain number of generations (or computation time), it is possible to say with high confidence that no feasible point exists; the second threshold point is defined by monitoring the expected number of generations to get a feasible solution, if one exists. Thus, if after a certain number of generations a feasible solution has not been obtained, it is likely that there is no acceptable solution. The higher the threshold on the number of generations, the smaller is the chance of rejecting marginally good components. However, the maximum number of generations is limited by the computation time acceptable for on-line inspection. There needs to be an acceptable trade off between the maximum computation time and the chances of false negative reports.

For comparison, tests with the GRG method implemented by GINO for the same data sets required a maximum of 15 iterations, for the second model problem, to get a feasible solution or to detect infeasibility. The computation time was no more than 10 seconds on 386 PC computer, 20 MHz clock. The most distinct advantage of this method over genetic search is in its more definitive indication of the feasibility condition. For the problems studied here there is no decisive computation time advantage of one method over the other. As the introduction of a small buffer zone largely removes the ending difficulty with only small chances of false results in marginal cases, the choice between the two methods is finely balanced. However, there is evidence that as the complexity of problems (number of parameters and constraints) increases, the computational effort of genetic search rises much less rapidly than that of formal methods so it may become increasingly attractive for full inspection of workpieces. For the present, we conclude that it remains a promising candidate for some classes of automated inspection.

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- [2] A more practical method based on Hook's law is proposed, and the measured coordinate data is also successfully corrected by this method.
- [3] The optimum clamping position of a thin plate which minimizes deformation due to the measuring force and gravity is investigated.

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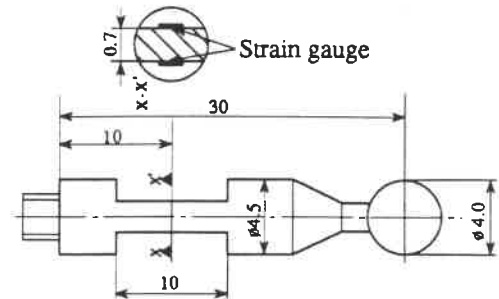


Fig.1 Stylus dimensions

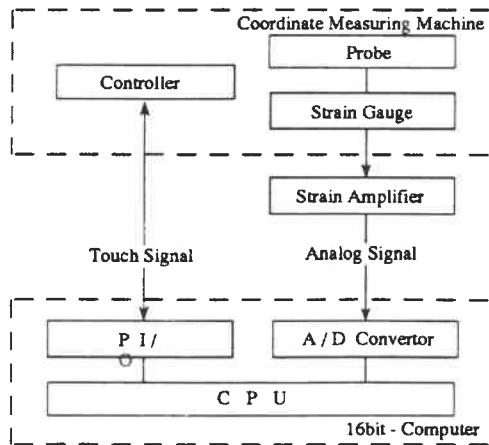


Fig.2 3D-CMM computer system connection

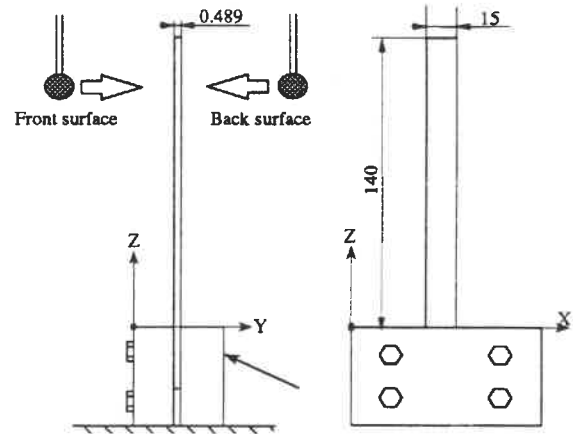


Fig.3 Beam model for measuring test

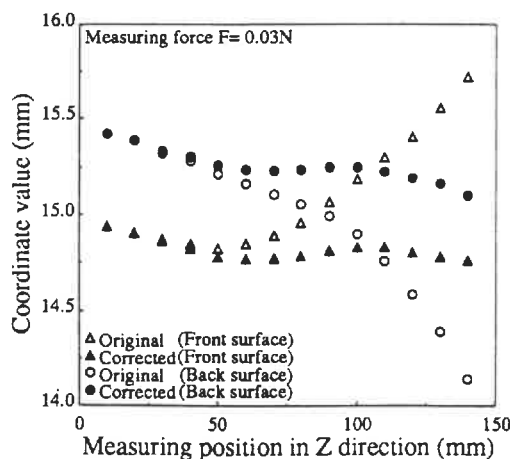


Fig.4(a) Measurement results (correction by FEM)

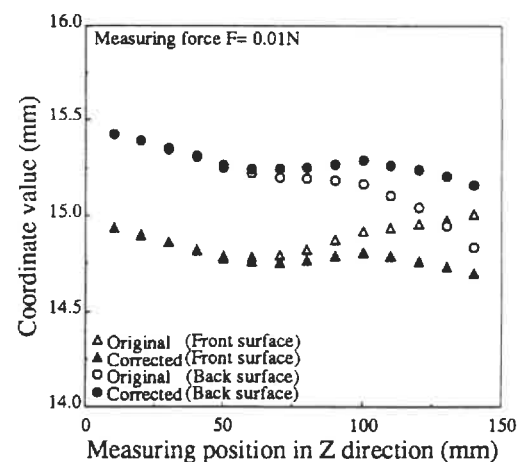


Fig.4(b) Measurement results (correction by FEM)

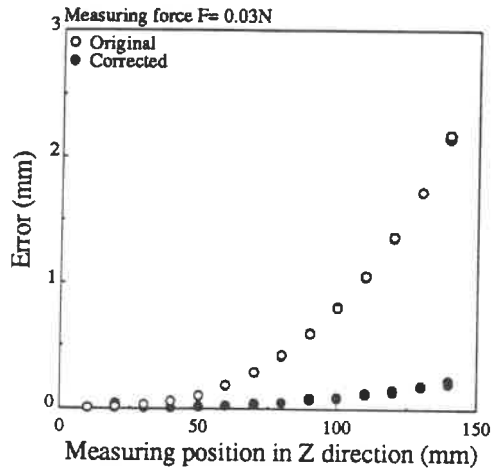


Fig.5(a) Measuring error
(correction by FEM)

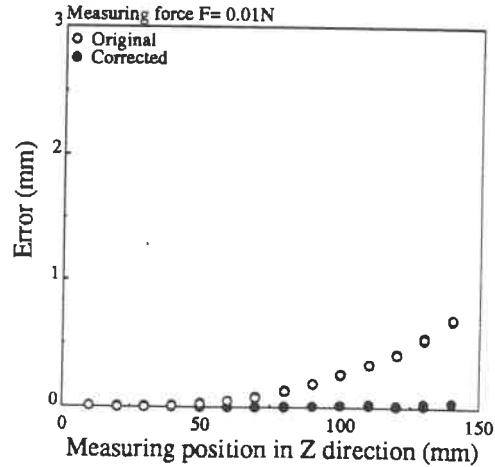


Fig.5(b) Measuring error
(correction by FEM)

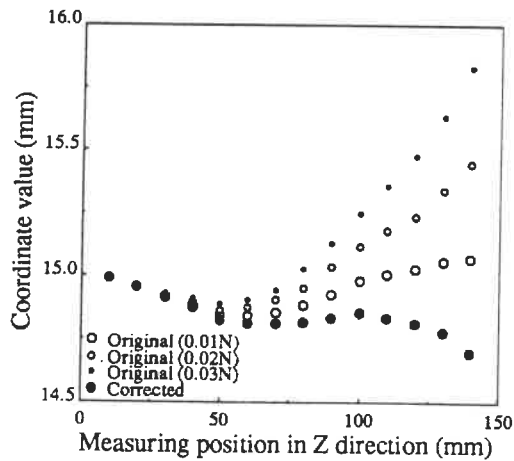


Fig.6(a) Measurement results
(Front surface)
(correction by Hook's law)

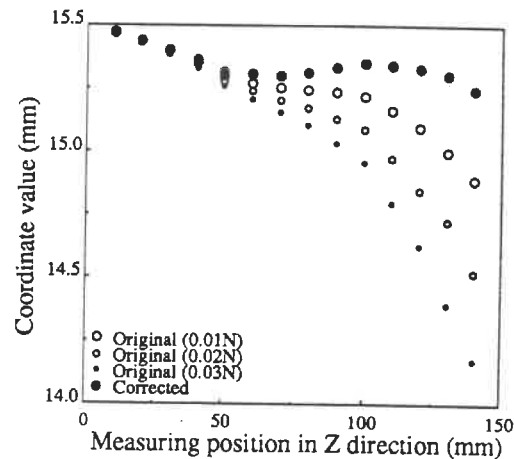


Fig.6(b) Measurement results
(Back surface)
(correction by Hook's law)

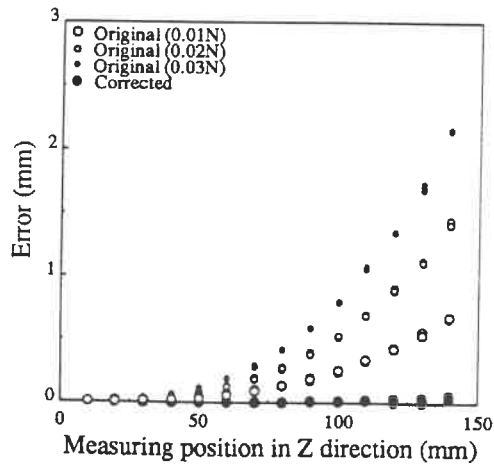


Fig.7 Measuring error
(correction by Hook's law)

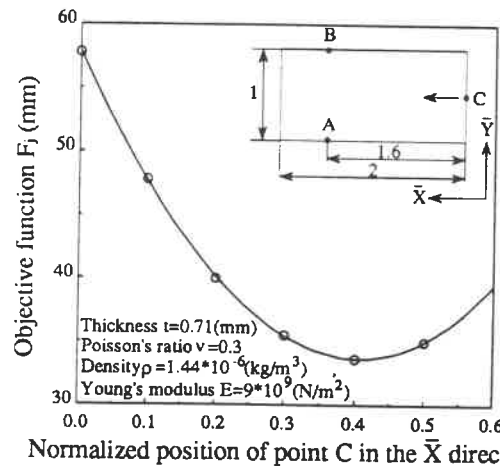


Fig.8 Evaluation by objective function F_j

Angle Measurement based on the Internal-Reflection Effect and its Application in the Measurement of Geometric Errors of Machine Tools

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1. Introduction

A new optical method of angle measurement, Angle Measurement based on the Internal-Reflection Effect (AMIRE), has recently been developed with such features as high resolution, compact size and low cost [1]. This paper describes a further development of this method with a multiple reflection scheme introduced to increase measurement sensitivity and its application in the simultaneous measurement of geometric errors of machine tools.

2. AMIRE with Multiple Reflections

AMIRE makes use of the characteristic that a light beam internally reflected from an air-glass interface in the vicinity of the critical angle undergoes a rapid reflectance change with the angle of incidence. The relation between the reflectance and the angle of incidence is inherently nonlinear. To find a linear relation, a differential detection method has been proposed and a new variable, the linearized reflectance R_l , has been defined as follows:

$$R_l(\Delta\theta) = \frac{R(\Delta\theta) - R(-\Delta\theta)}{R(\Delta\theta) + R(-\Delta\theta)}$$

$$= b_1(\theta_0)\Delta\theta + b_3(\theta_0)\Delta\theta^3 + \dots,$$

where θ_0 is the initial angle of incidence and $\Delta\theta$ is the angular displacement from θ_0 . $R_l(\Delta\theta)$ is a

odd-term-only polynomial of $\Delta\theta$ with its coefficients being functions of θ_0 . It has been shown that if θ_0 is assigned certain values (optimal angles), the sum of the higher order terms of $R_l(\Delta\theta)$ approaches zero and $R_l(\Delta\theta)$ becomes approximately linear with $\Delta\theta$. With a linear relation, $\Delta\theta$ can be readily measured with a high accuracy by detecting $R_l(\Delta\theta)$.

To improve the measurement sensitivity, a multiple reflection scheme has been proposed that uses elongated critical angle prisms as the detection prisms (Figure 1). Analysis showed that the measurement sensitivity is increased approximately with the square of the number of reflections, which implies that increasing the number of reflections is an effective way of increasing the measurement sensitivity. As in the

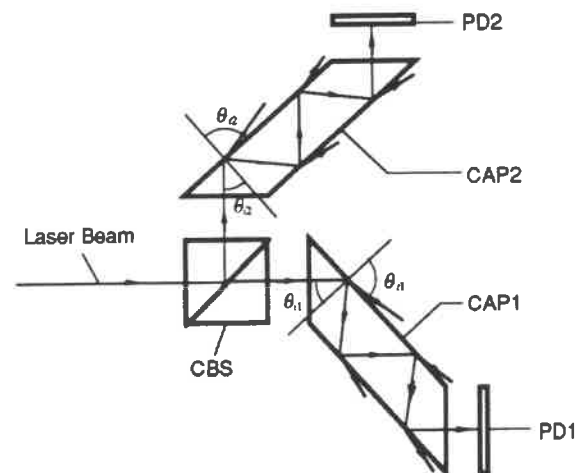


Figure 1 AMIRE with multiple reflections.

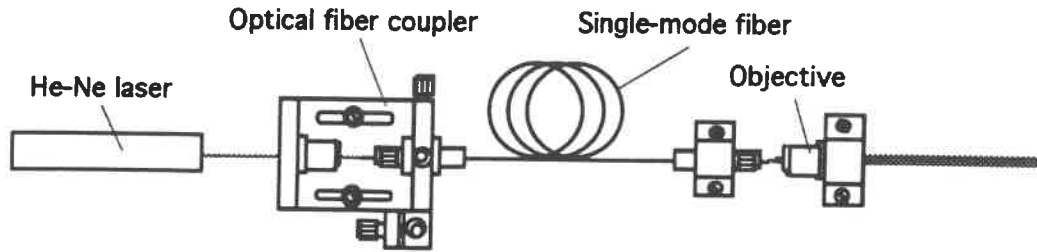


Figure 2 Single-mode fiber beam delivery system.

case of single reflection, the nonlinearity error can be minimized by setting θ_0 at certain optimal values which vary with the number of reflections. Since the requirement for most applications in machine tools is basically high resolution, this multiple reflection scheme should usually be adopted.

3. Multi-Degree-of-Freedom Geometric Error Measurement System (M-GEMS)

For a measurement system based on the laser alignment technique, the pointing stability of the laser beam is one of the most important factors that influence the measurement accuracy. In this research, a single-mode fiber beam delivery system, as illustrated in Figure 2, was used to improve this pointing stability. Experimental results showed that by using this fiber system, the pointing stability of a laser beam can be improved by approximately two orders of magnitude.

Figure 3 shows the principle of the simultaneous measurement of straightness and angular errors of a linear motion. The optics is essentially the same as that of an angle measuring sensor except that the photodiodes are replaced by PSDs. Figure 3 (a) shows the case when the moving object which carries the optics

has a straightness error. For simplicity, the diagram is drawn with the laser beam laterally shifted due to the straightness error instead of the moving object, which will not change the nature of the problem. As can be seen, the two PSDs give exactly the same position signal that represents the straightness error. Figure 3 (b) shows exactly the same optics but in the case of angular error. By using the intensity signals of the PSDs, this optics becomes purely an angle measuring sensor. Since the intensity signal is independent of the position of the beam spot and the position signal is independent of the intensity of the light beam, these two measurements are essentially independent of each other. If we denote x_{11} and x_{12} as the outputs of the first PSD and x_{21} and x_{22} as the outputs of the second PSD, we can calculate the straightness and angular errors as

$$s = \frac{(x_{11} - x_{12}) + (x_{21} - x_{22})}{(x_{11} + x_{12}) + (x_{21} + x_{22})},$$

$$\theta = \frac{(x_{11} + x_{12}) - (x_{21} + x_{22})}{(x_{11} + x_{12}) + (x_{21} + x_{22})}.$$

Figure 4 shows the optical layout of a multi-degree-of-freedom geometric error measurement system that simultaneously measures four error components (pitch, yaw, and two straightness errors) of one axis motion. It basically consists of a beam splitter plus two

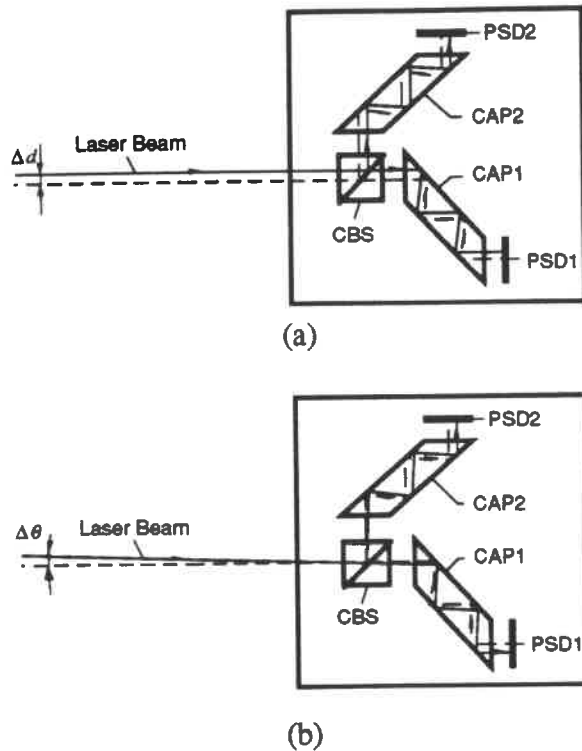


Figure 3 Optical layout for the simultaneous measurement of straightness and angular errors: (a) straightness error measurement; (b) angular error measurement.

identical optical systems shown in Figure 3, each lying in the horizontal or vertical plane. The signal conditioning electronics are assembled together with the optics to reduce signal transmission noise. The overall size of the system measures 114x76x57 mm.

4. Experiments

The system was calibrated by using the laser interferometer HP5528A. The results showed a nonlinearity of less than 0.3% for both straightness and angular error measurements. The drift of the system within an hour was found to be less than 0.1 μm and 0.1 arcsec for straightness and angular error measurements respectively.

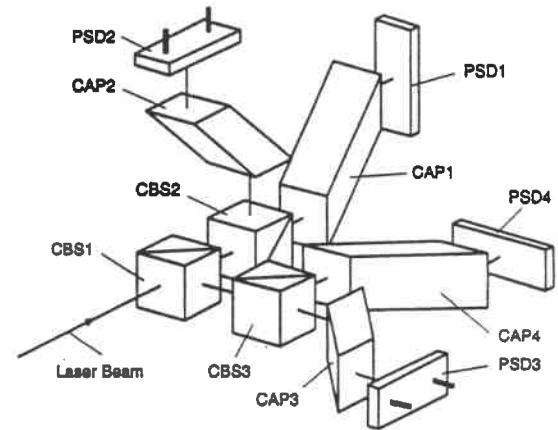


Figure 4 Optical layout of M-GEMS.

The system was used to measure the geometric errors of a mechanical slide which has a travel distance of approximately 300 mm and is driven by a DC motor. For comparison, HP5528A was used to measure the geometric errors at the same time. Since HP5528A can only measure one error component at a time, the setup needs to be changed for each measurement. However, no such change is needed for the developed system and all the four error components are measured simultaneously. Figure 5 shows the results. The overall agreement between the measurement results by the two systems can be seen. For straightness measurement, the results are not exactly the same because the measurement positions for the two systems were different due to the space limitation.

5. Conclusions

Following conclusions can be drawn from the research:

- 1) The measurement sensitivity of AMIRE with

multiple reflections is approximately proportional to the square of the number of reflections. Thus, increasing the number of reflections can greatly increase the measurement sensitivity.

2) The single-mode fiber beam delivery system can greatly improve the beam pointing stability of a laser beam so as to provide a highly accurate measurement reference.

3) The developed system for simultaneous measurement of geometric errors is a high

resolution, compact and low cost system with a measurement resolution of better than $0.1\text{ }\mu\text{m}$ for straightness measurement and better than 0.1 arcsec for pitch and yaw measurement.

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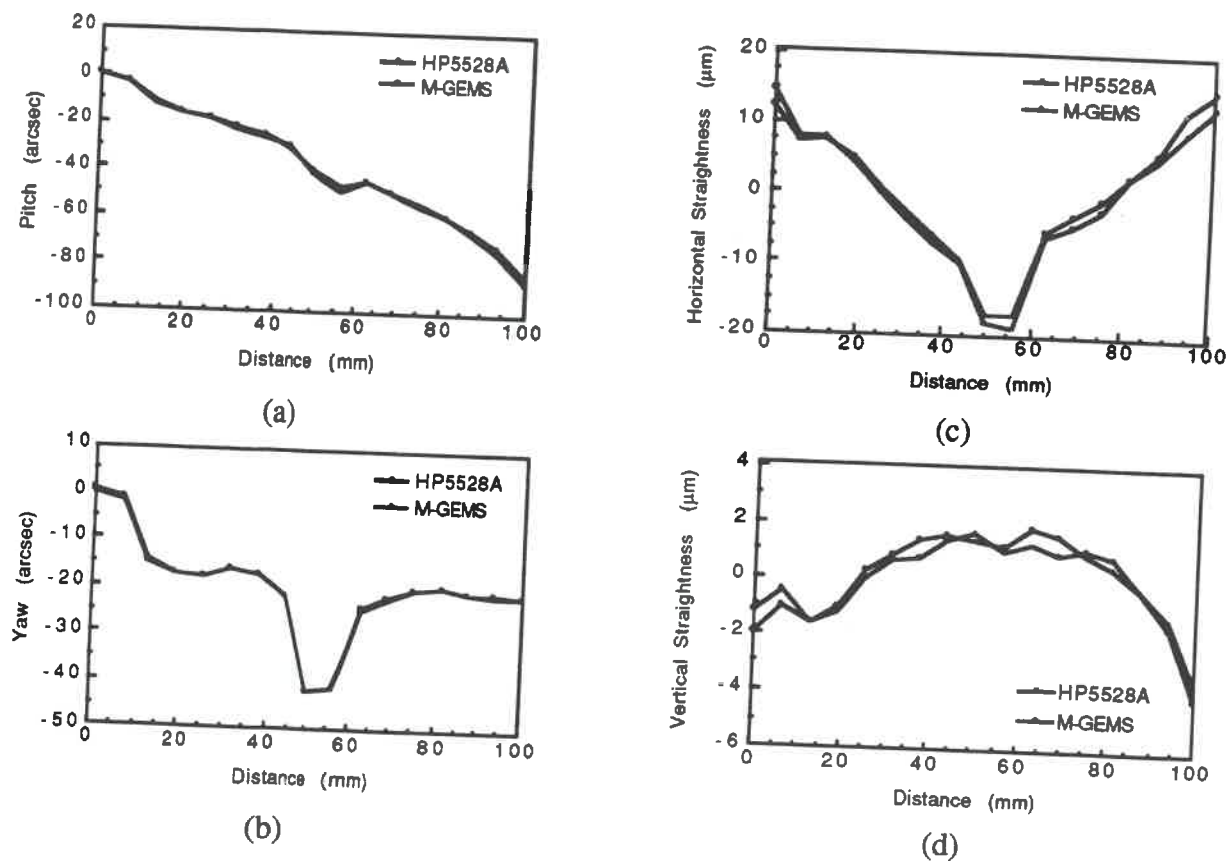


Figure 5 Measurement results: (a) pitch; (b) yaw; (c) horizontal straightness; (d) vertical straightness.

In the above equations, when the number of the independent equations is equal to the element number of vector $\hat{x}$, unknown vector x (or the circularity profile and error motion) can be determined by solving the simultaneous linear equations. When the number of the equations is over the element number, vector x should be determined in the sense of the least squares as follows:

$$D^T \hat{y} = D^T D x.$$

In this method, the measuring point interval is $2\pi/t$ and the minimum interval between the sensors is $\text{Min}[n_1, n_2, \dots, n_{r-1}] \cdot 2\pi/t$. If constants $n_1, n_2, \dots, n_{r-1}$ are selected so that $\text{Min}[n_1, n_2, \dots, n_{r-1}]$ may be over 2 and as large as possible, the measuring point interval can be shortened compared with the minimum sensor interval.

Numerical simulation

The verification of this method has been made through numerical simulation for $r=3, 4$ and 5. Constants $n_1, n_2, \dots, n_{r-1}$ are selected as shown in Table 1. Such ratios of the constants as $n_1:n_2=2:4$ for $r=3$ and $n_1:n_2:n_3=2:2:6$ for $r=4$ cannot be adopted, because those make matrix D or $D^T D$ singular. In the simulation, it is assumed that the circularity profile and error motion have the following sinusoidal wave form,

Circularity profile: $y_i = A_y \sin(2\pi f_y i / t + \phi_y),$

Translational component of error motion: $d_i = A_d \sin(2\pi f_d i / t + \phi_d),$

Angular component of error motion: $k_i = A_k \sin(2\pi f_k i / t + \phi_k), \quad i = 0, 1, \dots, t-1.$

The amplitude, frequency, phase and other conditions examined are shown in Table 2.

Table 1 Combinations of constants $n_1, n_2, \dots, n_{r-1}$

r=3		r=4						r=5			
n_1	n_2	n_1	n_2	n_3	n_1	n_2	n_3	n_1	n_2	n_3	n_4
3	2	2	2	3	3	4	3	3	3	4	5
4	3	2	3	2	2	2	7	3	3	5	4
5	2	2	3	3	2	3	6	3	4	3	5
5	3	3	2	3	2	4	5	3	4	4	4
5	4	2	2	5	2	5	4	3	4	5	3
7	2	2	3	4	2	6	3	3	5	3	4
7	3	2	4	3	3	7	2	4	3	4	4
6	5	2	5	2	3	2	6	3	4	5	3
7	4	3	2	4	3	3	5	4	3	3	5
8	3	2	3	5	3	4	4	4	3	3	5
9	2	2	5	3	3	5	3	4	3	4	4
		3	2	5	4	2	5				
		3	3	4	4	3	4				

The amplitudes of the profiles calculated from the simulated distance data $\hat{y}$ were compared with true values using Fourier analysis. As the result, it was found that the measurement error is below 0.1% for all conditions mentioned above. The influence of disturbance was also analyzed in the simulation. Figure 3 shows the relation between the error in profile

Table 2 Conditions of simulation

Diameter of objective	100mm
Number of measuring points t	128, 256
A_y, A_d, A_k	0, 1, 2, 5 μm
f_y, f_d, f_k	2, 3, 5, 10, 15
ϕ_y, ϕ_d, ϕ_k	0, $\pi/4, \pi/2$ rad

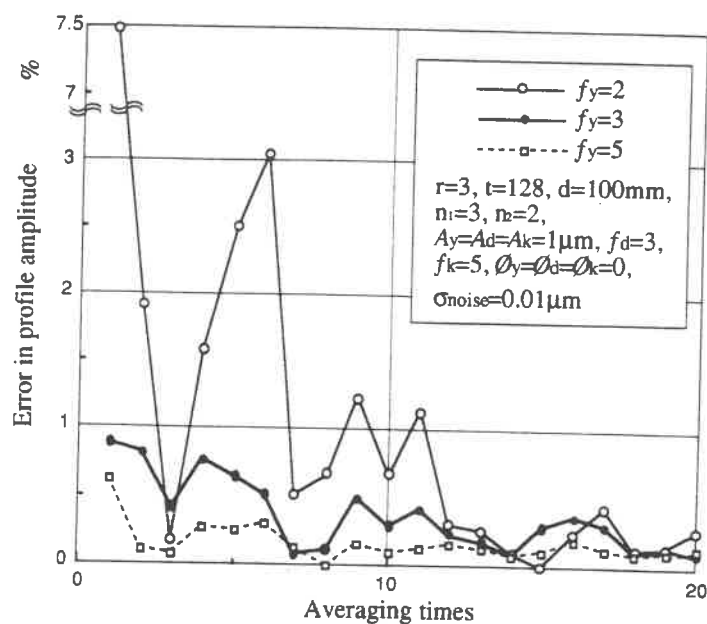
amplitude and the number of averaging times, when data y^* are contaminated with Gaussian random noise of standard deviation $\sigma=0.01\mu\text{m}$. It can be seen from this figure that the disturbance significantly reduces the measurement accuracy in the case of the smaller f_y and r .

Conclusions

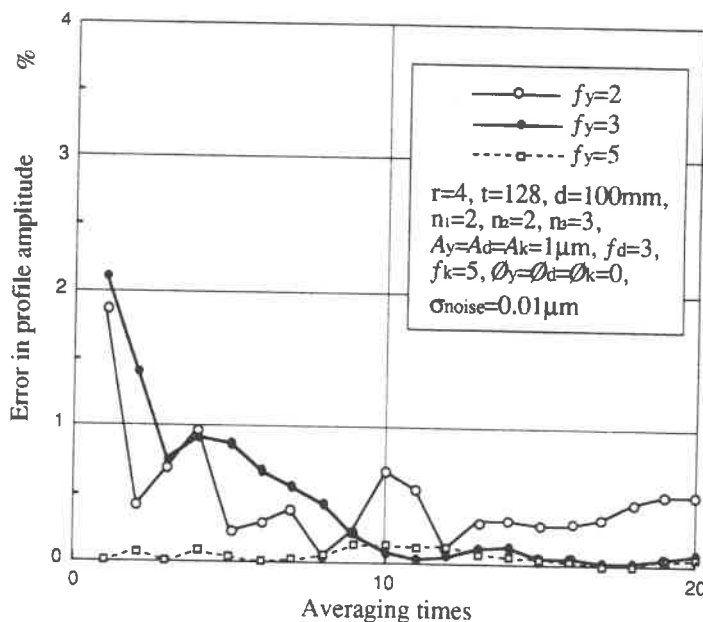
A serial multi-point method for measuring the circularity profile with a small step angle has been proposed and the measurement accuracy has been discussed through numerical simulation.

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(a) Serial 3 point method



(b) Serial 4 point method

Fig.3 Influence of disturbance on measurement accuracy

ESTIMATION METHOD FOR ALIGNMENT ERRORS OF MACHINES TOOLS BASED ON MACHINED PART SHAPE MEASURING

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1. Introduction

In this paper, an estimation method for alignment errors of machine tools based on machined part shape measuring is proposed. The proposed method consists of 1) mathematical representation of form shaping function of machine tools which is originally proposed by V.T.Portman, 2) analysis of influence of alignment errors on the machined errors, 3) identification procedure of alignment errors of machine tools based on off-line measuring, and 4) computer simulation and results evaluation.

2. Representation for Form Shaping Function of Vertical Milling Machine

Vertical milling machine consists of x, y and z axis translating driving units and spindle rotating driving unit as shown in Figure 1. According to the theory of V.T.Portman, the form shaping function of the 3-Axis vertical milling machine is given by Equation 1.

$$\mathbf{r}_0 = \mathbf{A}^1(x)\mathbf{A}^2(y)\mathbf{A}^3(z)\mathbf{A}^6(\theta)\mathbf{r}_t \quad (1)$$

where, $\mathbf{A}^1(x)$, $\mathbf{A}^2(y)$, and $\mathbf{A}^3(z)$ are translation matrix for x, y and z axis respectively, $\mathbf{A}^6(\theta)$ is rotation matrix around the z-axis, $\mathbf{r}_t$ is a vectorial representation of cutting tool edge, and $\mathbf{r}_0$ is the trajectory of motion of tool point relative to the workpiece.

3. Form Shaping Function of Machine Tools with Alignment Errors

Alignment error ε_i between two units is able to be represented by the following generalized error matrix, neglecting the elastic deformation of units.

$$\varepsilon_i = \begin{bmatrix} 0 & -\gamma_i & \beta_i & \delta_{xi} \\ \gamma_i & 0 & -\alpha_i & \delta_{yi} \\ -\beta_i & \alpha_i & 0 & \delta_{zi} \\ 0 & 0 & 0 & 0 \end{bmatrix} \quad (2)$$

where, δx_i , δy_i , δz_i are error components of translation regarding x, y, z axis, and α_i , β_i , γ_i are error component of rotation around x, y, z axis.

Deriving procedure of the form shaping function of the machine tools with alignment errors is shown in Figure 2. The positioning error of points of machined surface by the alignment error is given by Equation 3.

$$\begin{aligned} \mathbf{r}(x, y, z, \theta) = & \mathbf{A}^1\mathbf{A}^2\mathbf{A}^3\mathbf{A}^6\mathbf{r}_t + \varepsilon_0 \mathbf{A}^1\mathbf{A}^2\mathbf{A}^3\mathbf{A}^6\mathbf{r}_t + \mathbf{A}^1\varepsilon_1\mathbf{A}^2\mathbf{A}^3\mathbf{A}^6\mathbf{r}_t \\ & + \mathbf{A}^1\mathbf{A}^2\varepsilon_2\mathbf{A}^3\mathbf{A}^6\mathbf{r}_t + \mathbf{A}^1\mathbf{A}^2\mathbf{A}^3\varepsilon_3\mathbf{A}^6\mathbf{r}_t + \mathbf{A}^1\mathbf{A}^2\mathbf{A}^3\mathbf{A}^6\varepsilon_4\mathbf{r}_t \end{aligned} \quad (3)$$

4. Model of Cutting Tool and Machined Surface Representation

To apply the same method for modeling of a cutting tool, we obtain Equation 4 for a end-mill cutter and Equation 5 for a facing cutter with respect to the origin of the cutting

tool coordinate.

$$r_t(z_T) = A^6(2\pi z_T / P) A^3(z_T) A^1(R) e^4 \quad 4)$$

$$r_t = A^6(a_i) A^1(R) e^4 \quad 5)$$

Substituting Equations 4 and 5 into Equation 3, we can obtain the form shaping function for the vertical milling machine with the end-mill or the face mill.

5. Generation of Machining Surface Shape

Figure-3 shows the deriving procedure for a machining shape from the form shaping function. Machining shape can be derived under the following constraints.

Enveloping constraints: are the conditions of mutual perpendicularity of the velocity of the cutting point and the normal vector of the machined surface.

Functional constraints: are to specify the driving ratio among the velocity of motion units or specifying the position of unit for obtaining the required shape.

6. Relationship between Alignment Errors and Machined Surface Errors

A vertical milling machine has 30 alignment error components as shown in Table-1. Effects of error components are classified into the following three groups, 1) positioning deviation, 2) angle deviation and 3) form deviation. Here, we assume that measuring of machined part is done at a post process measuring using CMM. Then we can not detect the error components which give an effect to positioning deviation. And also, we assume that there are no alignment errors between x-table and workpiece, and spindle and cutting tool, because it is easy to identify it by using suitable jigs.

7. Identification of Alignment Error Components and It's Value

In Table-1, we can recognize that the alignment error components which affect angle deviation and form deviation are $\alpha_2, \alpha_3, \beta_1, \beta_2, \beta_3$, and γ_1 . Error components α_2 and α_3 give a same effect on machined surface, and β_1, β_2 , and β_3 are also same. Then we introduce new two parameters $\alpha (= \alpha_2 + \alpha_3)$, $\beta (= \beta_1 + \beta_2 + \beta_3)$. Finally the problem becomes to identify three error components. Deriving procedure for values of these alignment error components is shown in Figure 4. At first, to identify values of α and β , we derive the relationship equation between the alignment error components and the machined surface representation equation in facing X-direction facing surfaces become to a quadric surface as given by Equation 6, when alignment errors α, β are exist.

$$Y^2 + \frac{\alpha^2 + \beta^2 + 1 + \sqrt{(\alpha^2 + \beta^2 + 1)^2 - 4\beta^2}}{\alpha^2 + \beta^2 + 1 - \sqrt{(\alpha^2 + \beta^2 + 1)^2 - 4\beta^2}} Z^2 = \frac{2\beta^2 R^2}{\alpha^2 + \beta^2 + 1 - \sqrt{(\alpha^2 + \beta^2 + 1)^2 - 4\beta^2}} \quad 6)$$

On the other hand, from the measuring data set on the machined surface using CMM, we can derive the following quadric surface equations using the Least Square Method for the X-direction facing surface .

$$Y^2 + K_{11} Z^2 - K_{12} = 0 \quad 7)$$

From the combination of Equation 6 and Equation 7, we can derive the values of alignment error components (α, β) as given by the following equations

$$\alpha = \pm \frac{\sqrt{(R^2 - K_{12})(K_{12} - K_{11}R^2)}}{\sqrt{K_{11}R^2}} \quad 8)$$

$$\beta = \pm \frac{K_{12}}{\sqrt{K_{11}R^2}} \quad 9)$$

Secondly, using the derived values of (α, β) , and verification of perpendicularity between X-table and Y-table, we can derive the value of γ_1 . For verify the perpendicularity of X-table and Y-table, let consider the end mill cutting of a plane which is perpendicular to the x-y table. To solve Equations 3 and 4 under the functional constraints which are given by the following equations;

$$a_k X + b_k Y + c_k = 0 \quad 10)$$

$$z = C_1 \quad 11)$$

we can obtain Equation 12 for the plane and Equation-13 for the normal vector.

$$a_k X + (a_k \gamma_1 + b_k) Y + \{a_k(\alpha \gamma_1 - \beta) + b_k \alpha\} Z + d_k = 0 \quad 12)$$

$$n_k = \begin{bmatrix} \frac{a_k}{n_k} & \frac{a_k \gamma_1 + b_k}{n_k} & \frac{a_k(\alpha \gamma_1 - \beta) + b_k \alpha}{n_k} \end{bmatrix} \quad 13)$$

On the other hand, we can derive the following implicit plane equation from the measured data set on the machined surfaces($k=1, \dots, n$) using CMM

$$A_k X + B_k Y + C_k Z = 0 \quad 14)$$

From Equation 14, we can obtain the normal vector for k-th plane which is given by Equation 15.

$$m_k = \begin{bmatrix} \frac{A_k}{m_k} & \frac{B_k}{m_k} & \frac{C_k}{m_k} \end{bmatrix}^T, \quad m_k = \sqrt{A_k^2 + B_k^2 + C_k^2} \quad 15)$$

For two perpendicular planes(i-th and j-th), an inner product of the analytical normal vectors(Equation 13) should be equal to an inner product of measured based normal vectors(Equation 15), Then we can obtain the following Equation:

$$\begin{aligned} & a_i a_j \{ (1 + \gamma_1^2 + (\alpha \gamma_1 - \beta)^2) \} + b_i b_j (1 + \alpha^2)^2 + (a_i b_j + a_j b_i) \{ \gamma_1 + (\alpha \gamma_1 - \beta) \} \\ & = K_{ij} \sqrt{a_i^2 \{ (1 + \gamma_1^2 + (\alpha \gamma_1 - \beta)^2) \} + b_i^2 (1 + \alpha^2)^2 + 2 a_i b_i \{ \gamma_1 + (\alpha \gamma_1 - \beta) \} } \\ & \times \sqrt{a_j^2 \{ (1 + \gamma_1^2 + (\alpha \gamma_1 - \beta)^2) \} + b_j^2 (1 + \alpha^2)^2 + 2 a_j b_j \{ \gamma_1 + (\alpha \gamma_1 - \beta) \} } \end{aligned} \quad 16)$$

where, $K_{ij} = m_i \cdot m_j$. Substituting the values of α, β into Equation 16, we can obtain the value of γ_1 .

8. Conclusion

In this paper, we show the deriving procedure of alignment error components values for vertical milling machine from the measured data set on machined surface using CMM. The following results are obtained:

- 1) The relationship between the alignment errors of milling machine and machined errors which consists of location, geometry and size errors is derived.
- 2) In milling, facing, cylindrical and cone surfaces always became to quadric surface.

3) Estimation procedure of alignment errors of milling machine tools based on measuring of machined parts is proposed.

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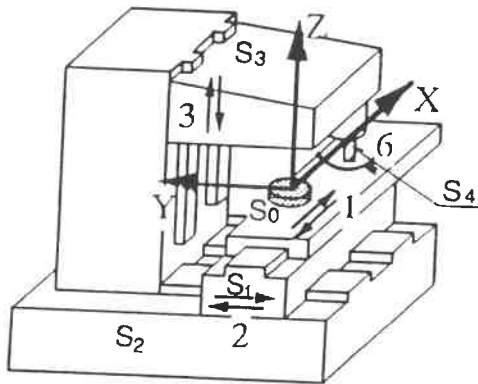


Fig. 1. Three Axis Milling Machine

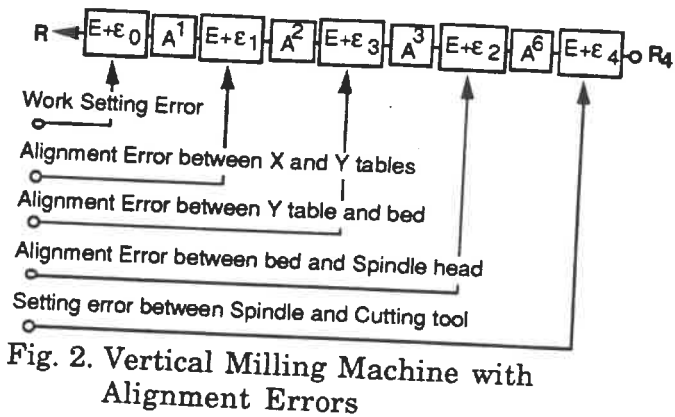


Fig. 2. Vertical Milling Machine with Alignment Errors

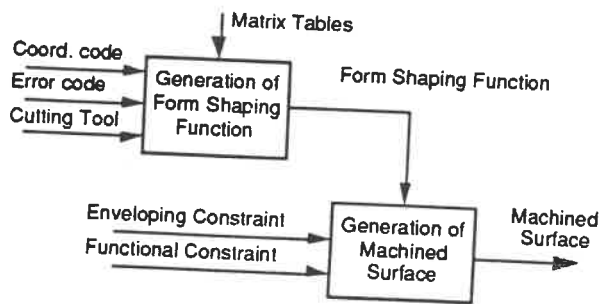


Fig. 3. Generation of Form Shaping Function and Machined Surface

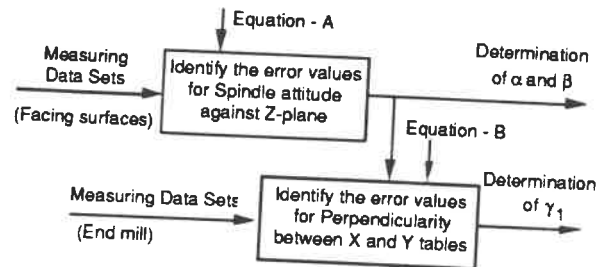


Fig. 4. Alignment errors estimation procedures

Table 1. Influences of Alignment errors component on machined surface

error	Tolerance		error	Tolerance		error	Tolerance		error	Tolerance		error	Tolerance		
	End mill	Face Cutter		End mill	Face Cutter		End mill	Face Cutter		End mill	Face Cutter		End mill	Face Cutter	
α_0	No	P	α_1	No	P	α_2	A	F	α_3	A	F	α_4	---	---	No: No effect P : Position F : Form A : Angle
β_0	No	P	β_1	A	F	β_2	A	F	β_3	A	F	β_4	---	---	
γ_0	No	No	γ_1	A	No	γ_2	No	No	γ_3	No	No	γ_4	---	---	
δ_{x0}	No	No	δ_{x1}	No	No	δ_{x2}	No	No	δ_{x3}	No	No	δ_{x4}	---	---	
δ_{y0}	No	No	δ_{y1}	No	No	δ_{y2}	No	No	δ_{y3}	No	No	δ_{y4}	---	---	
δ_{z0}	No	P	δ_{z1}	No	P	δ_{z2}	No	P	δ_{z3}	No	P	δ_{z4}	---	---	

Measurement of Strain Distribution in Diamond cut Surface Layer using Micro Grid Applied Electron Beam Lithography

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Introduction

Precision diamond turned surfaces had been studied mainly from geometrical viewpoints, such as the surface roughness, the dimensional accuracy and the mechanism of mirror surface cutting. On the other hand, recently great interest has been focused on not only geometrical characteristics of the machined surface but also properties of the subsurface layer. Especially, measurement of the residual stress and strain distribution in the subsurface layer have recently become an important subject of research for understanding the cutting mechanism. From these points of view, in this paper, a newly developed measuring method of strain distribution using electron beam lithography was applied to the orthogonal cutting of copper metal. Measurement of micro strain and its distribution in the subsurface layer became possible by this method. The dependence of subsurface strain on tool edge geometries was found out through the orthogonal cutting with a micro cutting device.

Micro Grid Fabrication for Strain Measurement

In the past, observation and measurement of the material deformation in cutting process were carried out with various methods, which was used scratch lines or micro indentations of hardness tester or specified grains in the substrate. It is difficult to measure microscopic deformation using these methods. Then we have developed a new method applied the electron beam lithography generally used a semiconductor micro process to the fabrication of $5\mu\text{m}$ square grids on the surface of metal specimen. Fig.1 shows the flow chart of this method. At first, the polished and etched oxygen free copper (OFC) surface was coated with photo resist and was exposed to the focused electron beam using a scanning electron microscope (SEM) which was modified able to change scanning speed and pitch arbitrary. Next developing the exposed photo resist, $5\mu\text{m}$ square resist array was remained. Then gold sputtering deposition were carried out on it. Finally dissolving fine square resist by the solution, gold micro grids array on the OFC surface in $400 \times 300\mu\text{m}$ square region which we use was formed. These fine grids can be clearly observed through a microscope especially with SEM because copper and gold have a difference of reflection rate to electron beam.

Orthogonal Cutting with Micro Cutting Device

Orthogonal diamond cutting were carried out for investigating microscopic deformation behavior in the cutting surface with the newly developed micro cutting device. Two types of

diamond flat tip were examined to clarify an influence of tool edge geometry on the strain distribution in a diamond cut surface layer. One is 0 deg. in rake angle and 5 deg. in clearance angle, and another is 0.2mm chamfered at the edge, in this case -47.5 deg. in rake angle. The tool is set as the rake face is perpendicular to cutting direction in the micro cutting device, shown in Fig.2. The micro cutting device makes it possible to machine the specimen with micro grids and observe the side view during cutting with SEM or through the optical microscope.

Copper specimen is fixed with the specimen holder on the slider table and is cut orthogonally by the diamond tool. The driving source of the slider table is a piezo electric actuator and a lever system. Depth of cut is set in submicron order step with the slide wedge positioning at the back of tool holder. Two piezo electric dynamometers in each perpendicularly mounted on the tool holder detect cutting forces. Relative displacement of each grids after cutting to original positions is measured by SEM photographs of the copper specimen. Then the strain distribution below the cutting surface in each normal and tangential direction is directly calculated conforming to the following formula.

$$\varepsilon_x = \frac{x_2 - x_1}{x_1} \quad \varepsilon_y = \frac{y_2 - y_1}{y_1}$$

where x_1, y_1 are the original distances between grids to each direction and x_2, y_2 are the same after deformation.

Result

Figure 3 and Figure 4 are SEM photographs of deformed micro grids in the case of using 0 deg. rake angle tool and negative rake angle tool respectively. In the former, deformation of micro grids in the subsurface appears toward cutting direction and many sliding lines are produced from the surface into the deeper area. In the latter case, the deformation of micro grids reaches deeper area. This suggests that the subsurface damage is strongly influenced by diamond tools in particular rake angle. Figure 5 and Figure 6 illustrate the strain tangential to cutting direction and their distribution respectively. The strain right under the surface is compressed and more than 30 percent. But the strain decreases gradually and the depth of strain field is just less than $30 \mu\text{m}$ under the condition which the depth of cut is $7.4 \mu\text{m}$. On the other hand, negative rake angle tool produces greater compression strain above 20 percent in deeper field more than 0 deg. rake angle tool does. Tool edge geometries change by the various factor especially tool wear in the course of machining. And there is a experimental result that the surface roughness is not necessarily proportional to tool edge sharpness. That is why we choose different shaped tools. These result indicates that surface integrity such as the subsurface damage depends on tool edge geometry. Figure 7 shows a SEM photograph of another experiment for observing a detail of sheared area by using micro grids after cutting. The cutting condition is $50 \mu\text{m}$ in depth of cut, 30mm/min. in cutting speed and using 0 deg. rake angle tool. Bending grids from the free surface A to the tool tip A' shows a shear plane or shear area. Upper part of A-A' deforms largely but the situation of

complex deformation is visible through the trace of micro grids. In addition, grain B just under the shear plane indicates the grain boundary slidings. Thus this method is also effective to visualize the cutting mechanism.

Conclusion

Newly developed method using micro grid applied electron beam lithography was applied to the measurement of strain just under the diamond cut surface. Through the experiment using different shaped tools, the distribution of strain had a difference in each and this indicated the surface integrity was largely affected by tool edge geometries especially rake angle.

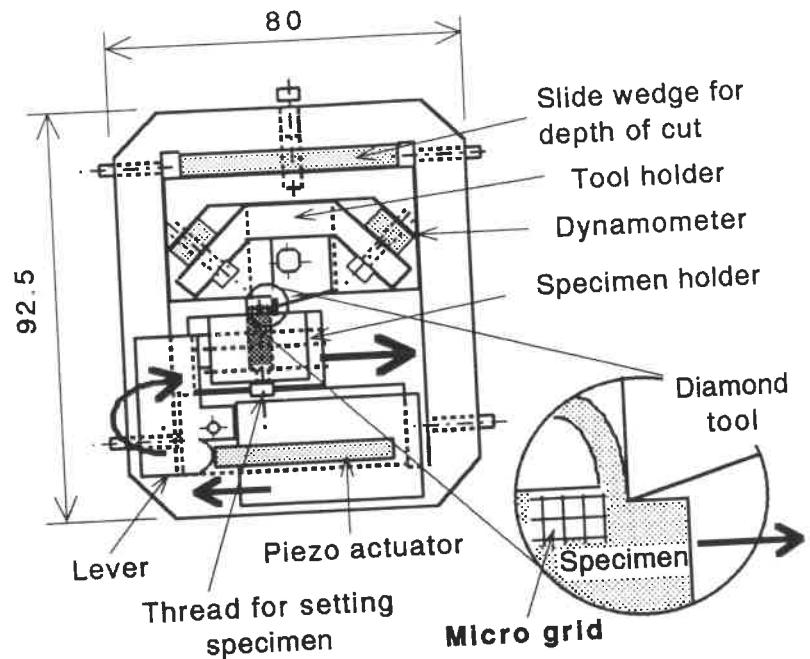
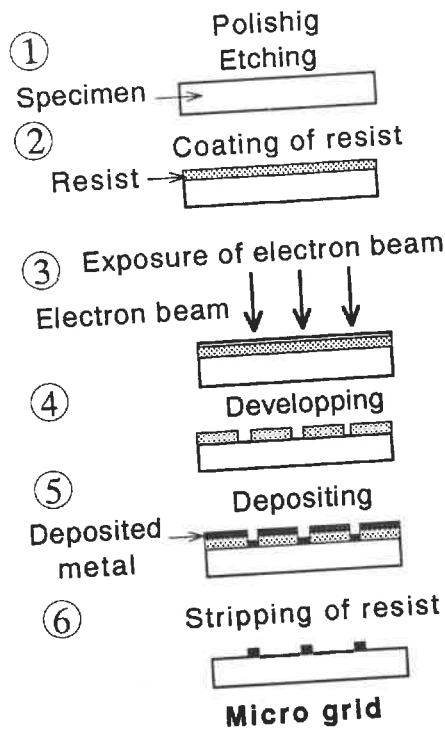


Fig.1 Flow diagram of Micro grid generation process

Fig.2 Schematic illustration of Micro cutting device

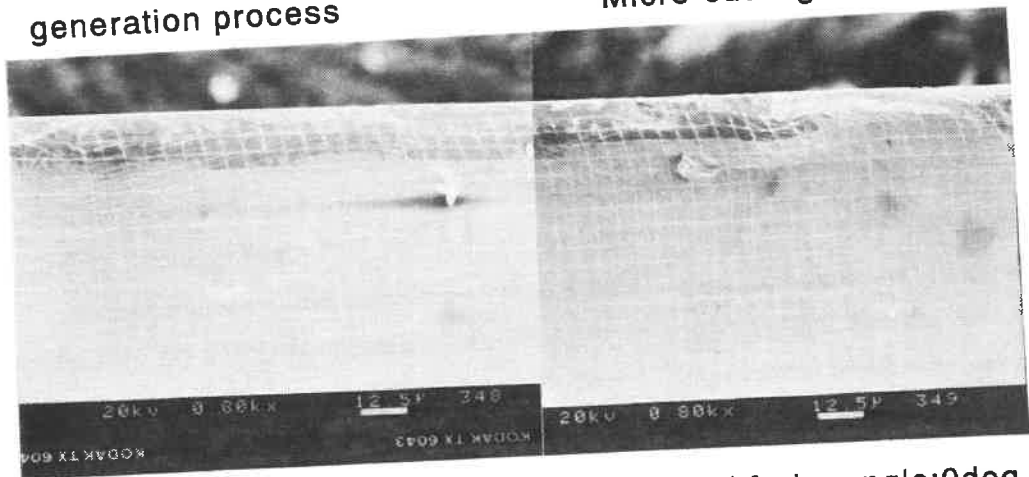


Fig.3 SEM photograph of deformed micro grid [rake angle:0deg., depth of cut:7.4um, cutting speed:0.8mm/min]

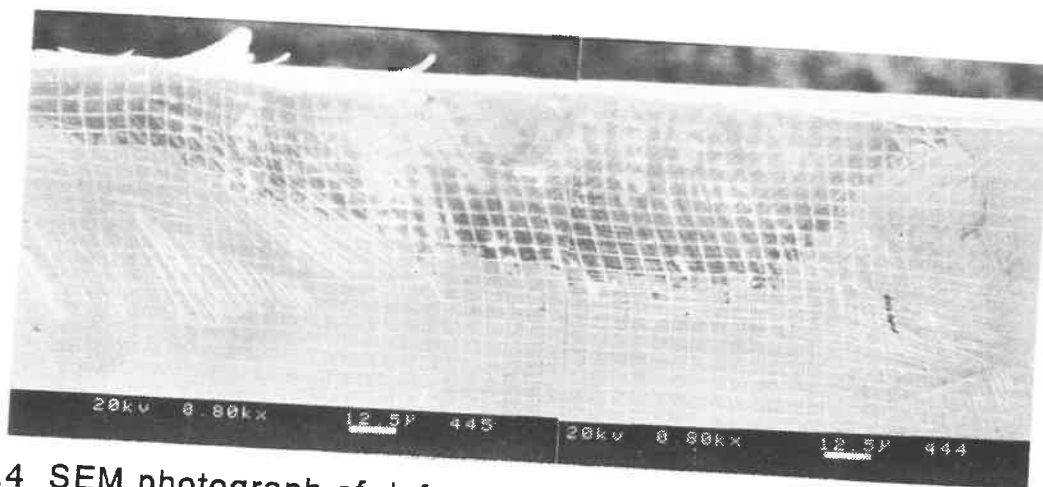


Fig.4 SEM photograph of deformed micro grid [rake angle:-47.5deg., depth of cut:8.8um, cutting speed:0.8mm/min]

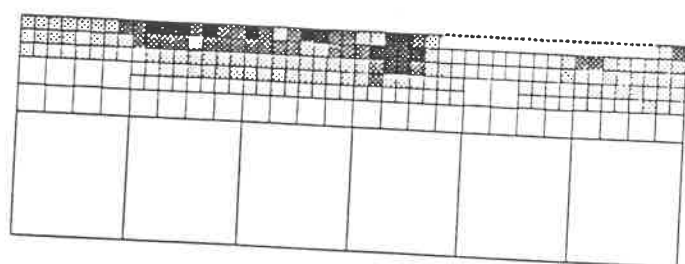


Fig.5 Distribution map of strain tangential to cutting direction [rake angle 0deg.]

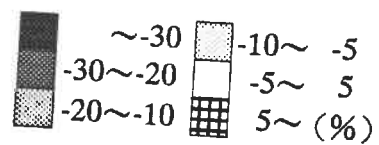
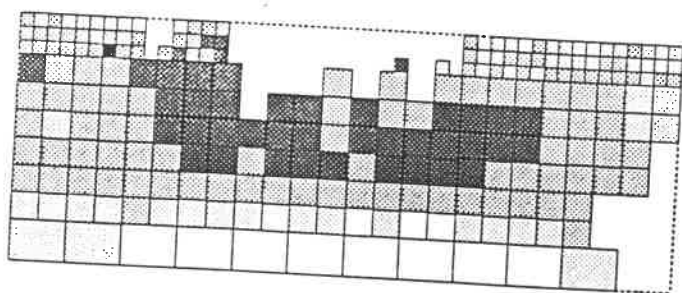


Fig.6 Distribution map of strain tangential to cutting direction [rake angle -47.5deg.]

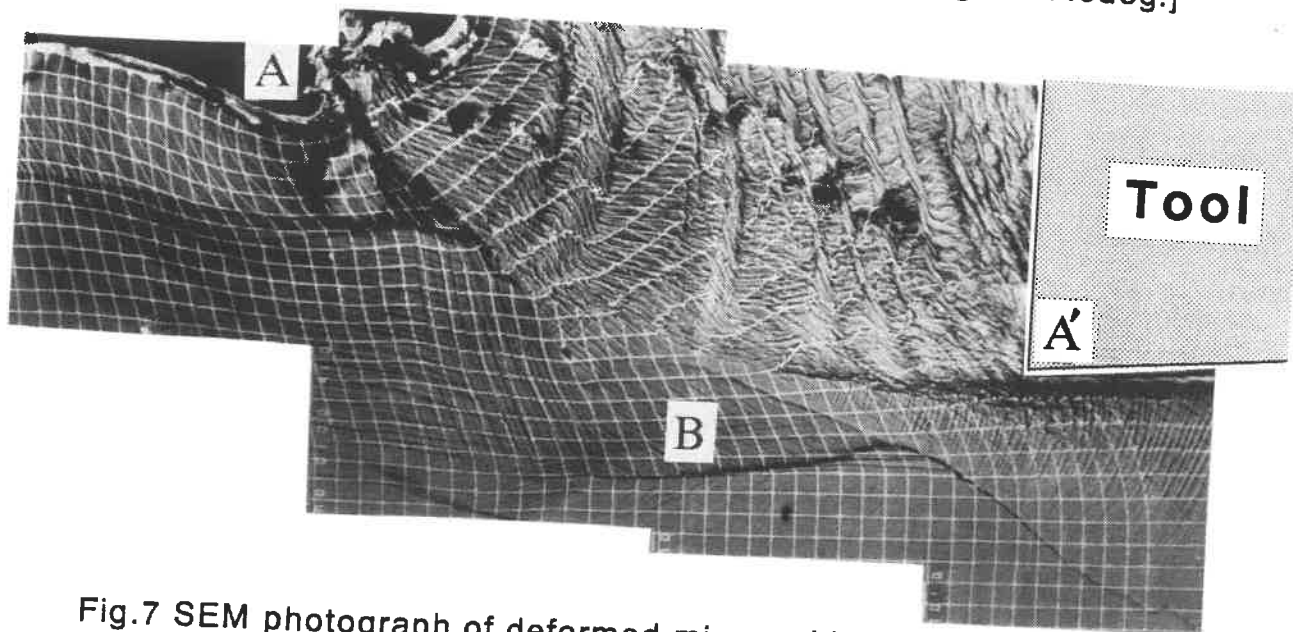


Fig.7 SEM photograph of deformed micro grid [depth of cut 50um]

UNCERTAINTY OF COORDINATE MEASURING MACHINE

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Summary

A vital element of flexible automation in a manufacturing system is the coordinate measuring machine (CMM). Flexibility, high accuracy, and different software support of these systems make up universal metrological systems.

This paper shows a very important parameter of CMM uncertainty. Namely, it presents an original approach to this problem, which is developed in our Laboratory for KOMEG's user members in our country.

The experimental platform is CMM UMC-850, with advanced software support.

Key words: CMM, Dimensional Metrology, Uncertainty.

1. PROBLEM DEFINITION

According to [1], length measurement uncertainty with CMM is defined as reliability of accurate measurement of distance between two points in parallel planes. The line that connects these two points is called the measuring line.

Measurement uncertainty is defined by terms: $|L_a - L_r| \leq u$, where: L_a -means measurement length obtained on CMM, L_r -nominal value of measuring length, and u -permissible value for measuring uncertainty, for 95% of measured values.

Length measurement uncertainty, which is being investigated in this paper, can be expressed by following equation:

$$U_i = A_i + K_i L \leq B_i$$

where: $i=1,2$ or 3 (one-dimensional, two-dimensional, and three-dimensional uncertainty of length measurement), A , K , B - constants, and L - length measurement.

The graphic illustration of this equation represents a diagram of length measurement uncertainty.

The constant B is defined by producers of CMM and it represents the parameter of accuracy of these systems.

2. CURRENT RESEARCH

CMM are complex engineering products, which have a complicated hardware and software structure. Their quality parameters (in production, installation and use) are defined by national or international standards. There are two approaches of testing length measurement

uncertainty.

The first is defined by a group of national and international standards or regulations of associations of CMM users [1-5]. These regulations are applied in testing measurement uncertainty of gauge block and laser interferometer.

The second approach applies metrological identification of characteristic metrological features through which length measurement uncertainty is defined. This approach also enables users of CMM to set parameters of measurement uncertainty on the basis of "etalon" measurement parts.

3. OUR APPROACH TO UNCERTAINTY RESEARCH

A CMM UMC-850 was installed in the Laboratory for Production Metrology and QA and after the assembly four groups of quality parameters were determined: (i) cinematic accuracy, (ii) measurement uncertainty, (iii) geometric axis accuracy, and (iv) measurement repeatability of "etalon" measurement parts. Values of parameters of length measurement uncertainty were the basis of this research [6,7].

Starting from the axiom-development of method and procedure for investigating measurement uncertainty, for KOMEQ equipment users club members in Yugoslavia for demo measurement parts, Fig.1, parameters for length measurement uncertainty have been defined.

For the same demo measurement parts, Fig.2, parameters of measurement uncertainty have also been defined on the basis of measuring tasks.

Parameters of length measurement uncertainty U_1 , U_2 and U_3 have been determined with 11 metrological tasks in CNC programme for these measurements. For testing measurement uncertainty by means of measuring tasks the number ranges from 3 to 7.

4. EXPERIMENTAL RESULTS

Obtained experimental results have been shown in Table 1 and 2. For demo measuring part seven characteristics have been measured under the following conditions: temperature of the measuring room -20 ± 0.1 °C and humidity of the room 45%. Probe tip diameter 5mm, length of probe bar 100mm, and probing force 0.1 N.

On the basis of measurement results, for parameter validation of measurement uncertainty, four parameters have been determined. Results from Table 1 and 2 are typical and refer to coordinate system of the machine: $x=204.2620$, $y=-904.0890$, and $z=-433.3121$.

By comparing obtained experimental results, with VDI standards, for example, the following can be concluded: (i) all tested parameters (for length measurement uncertainty and metrological tasks) are within permissible tolerance, (ii) for length measurement uncertainty the smallest dispersion has one-dimensional uncertainty, and the largest one three-dimensional uncertainty, (iii) the mean values for one and two-dimensional uncertainty of length measurement are approximately equal, (iv) with length measurement uncertainty on the basis of metrological tasks, the smallest dispersion has position and the biggest angularity, (v) mean values for perpendicularity and flatness are equal, while for position and angularity differ significantly, and (vi) all parameters, except two-dimensional uncertainty and position are in the second half of the permissible tolerance field.

These results show that the tested CMM has high parameters of measurement reliability.

5. FUTURE RESEARCH

Further study in this field will be conducted in two directions: (i) completion of presented metrology for testing repeatability, and (ii) further development of these approaches as Yugoslav standards in this field.

In this way the industry is enabled to test their CMM unidirectional.

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TAB.1.

	U1	U2	U3
Bi	145	230	315
min.	87.3	65	120
max.	127	145.7	241.8
mean (50)	105.2	107.1	174.2
% Ti	27.14	35.1	41.2

* all values are $\times 10^{-5}$

TAB.2

METROLOG. PARAM. TASKS	POSITION	PERPENDICULARITY	FLATNESS	ANGULARITY
Bi	230	315	315	315
min.	92	184	132	147
max.	175	235.4	246.1	286.4
mean (50)	101	208.4	191	226.7
% Ti	11.23	16.32	36.25	44.28

* all values are $\times 10^{-5}$

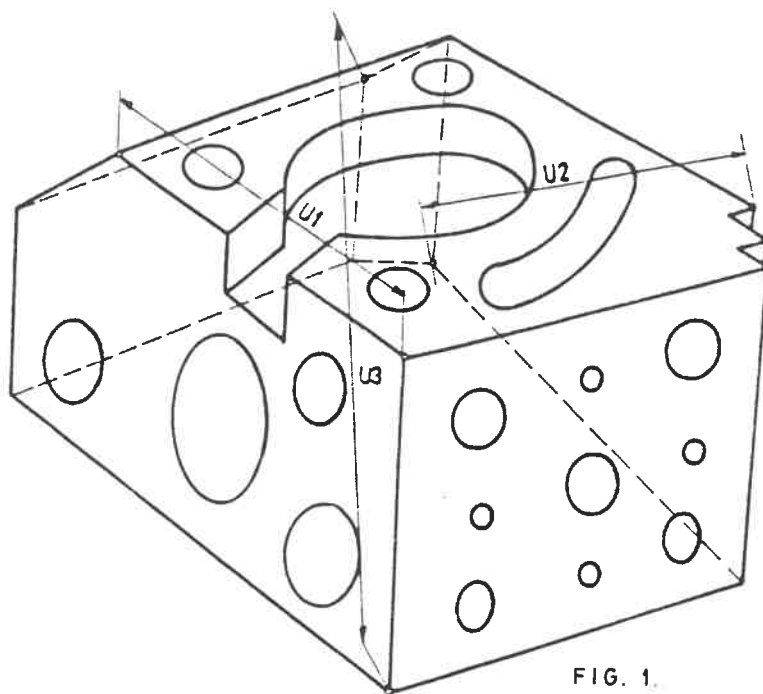


FIG. 1.

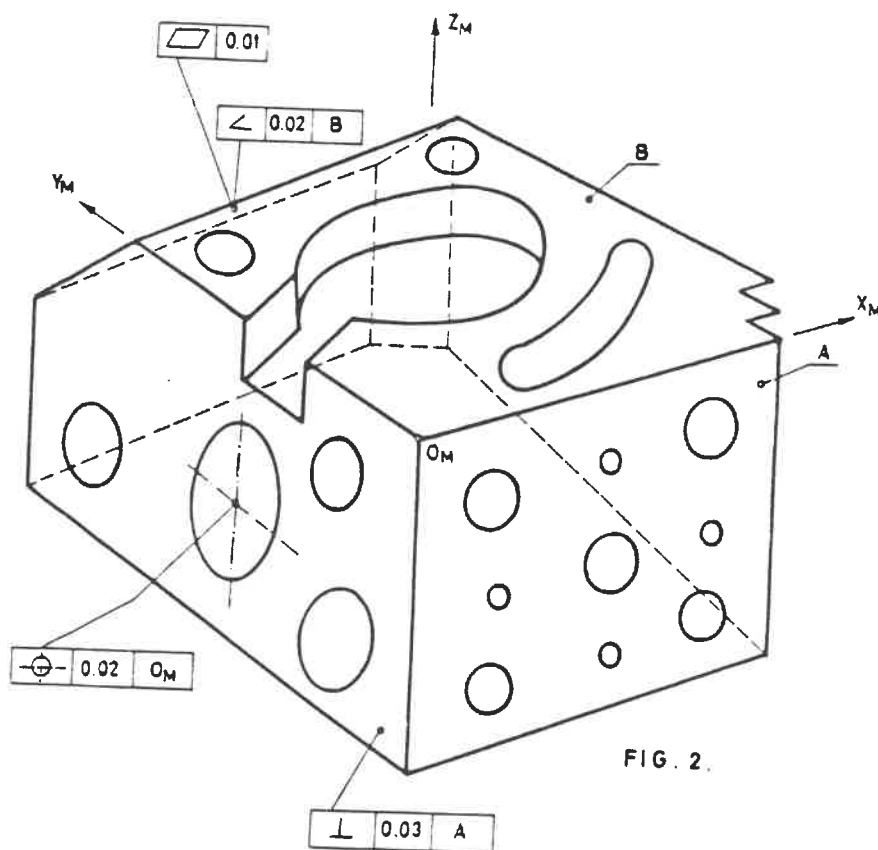


FIG. 2.

Non-contact 3-D Profile Sensor - Optical Ring Image Sensor -

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This paper describes development of a non-contact profile sensor which will be able to measure 3-D free form machined surface as well as 3-D model shapes made of clay or wood. The proposed profile sensor called "optical ring image sensor" has two major advantages. First, reflection light from a target surface in optical axis can be detected in all directions using a ring slit. Second, the radius of optical ring image is linearly proportional to the displacement of the surface in optical axis. It is concluded that this sensor makes it possible to measure the profiles within an accuracy of $\pm 10 \mu\text{m}$ for the 3-D free form surface with steep angles up to ± 80 degrees.

1. Introduction

Most 3-D profile sensors which have been developed until now are methods making contact with a sensor probe. Disadvantages of these techniques are (1) friction and elastic deformation occur, (2) it takes a long time to perform the measurement and (3) it is necessary to compensate for the radius of curvature of a sensor probe.

While a non-contact technique which measures the distance to a surface by use of the principle of triangulation is well known and this non-contact measuring method is very often used as a 3-D profile sensor. However, this technique has the following disadvantages: (1) a dead angle called "shadow effect" occurs and the displacement of the surface can not be correctly measured when the inclination angle of the surface to be measured is more than 45 degrees. (2) the location of a light spot to be detected is not linearly proportional to the displacement of the surface. The purpose of this paper is to develop a non-contact profile sensor which will be able to accurately measure 3-D free form machined surfaces as well as 3-D model shapes made of clay or wood without the disadvantages mentioned above. In this paper, the measurement principle of a non-contact profile sensor called "optical ring image sensor" is theoretically analyzed and the properties of this sensor are experimentally measured and discussed.

2. Principle of Measurement

The measurement principle of the proposed optical ring image sensor is shown in Fig.1. The laser beam is illuminated on the work surface at the distance of Z_1 from the focus length f_1 of object lens L_1 . Reflected light at the target surface passes through a ring slit of radius d and produces the temporary optical ring image of radius r at focus length f_2 of converging lens L_2 . The temporary optical ring image passing through the diffuse glass plate is magnified by the imaging lens L_3 of focus length f_3 located at the distance of a from the diffuse glass plate and is detected as the magnified optical ring image of radius R at the imageplane located at the distance of b from L_3 . First, the relationship between displacement Z_1 and ring radius r of the temporary optical ring image is derived by applying Newton's imaging formula to the first lens(L_1) and the second lens(L_2).

$$r = (d \cdot f_2 / f_1^2) Z_1 \quad (1)$$

Second, considering the imaging geometry with respect to lens L_3 , the diffused light of the temporary optical ring image emitted from the diffuse glass plate forms the magnified optical ring image by lens L_3 . Using general lens formula, We obtain.

$$r = (a - f_3) R / f_3 \quad (2)$$

Rearranging by substituting Eq.(2) into Eq.(1), finally, the relationship between the displacement Z_1 and the ring radius R can be written as follows.

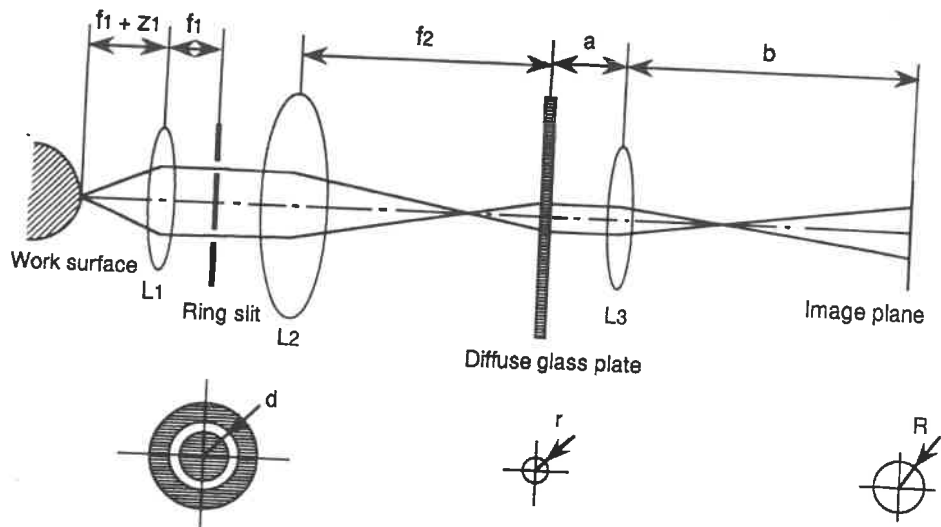


Fig.1 Geometry of optical ring image 3-D profile sensor

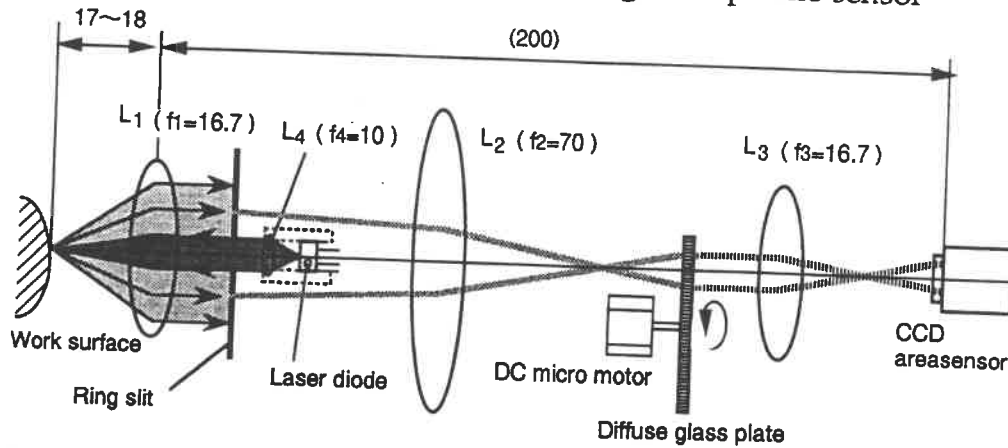


Fig.2 Configuration of the optical ring image 3-D profile sensor

$$R = \left(\frac{d \cdot f_2 \cdot f_3}{f_1 (a - f_3)} \right) Z_1 \quad (3)$$

The detected optical ring radius R is linearly in proportion to the work surface displacement Z_1 in the optical axis, where f_1 , f_2 , f_3 , d and a are constant values depending on the optical geometry.

3. Sensor System

Figure 2 shows the configuration of the sensor based on the measurement principle. It contains the basic optical system which forms the temporary optical ring image and the magnifying optical system which magnifies the temporary image. In addition, it includes the diffuse glass plate to be rotated by DC micro motor which is used for the reduction of the speckle noise. The proposed sensor consists of object lens, ring slit, converging lens, magnifying lens, laser diode, CCD area sensor etc. The ring slit with radius of $d=5\text{mm}$ is used to detect the reflected light from the work surface in all direction and is set at focal length ($f_1=16.7\text{mm}$) of lens L_1 in order to strictly satisfy the linearly proportional relationship of equation(10). Similarly, the diffuse glass plate is set at focal length ($f_2=70\text{mm}$) of lens L_2 . The magnifying lens with focal length of $f_3=16.7\text{mm}$ is located at distance of $a=25\text{mm}$ from the plate. The CCD areasensor with $13 \times 13 \mu\text{m}$ pixel size and 512×493 pixels is used for detecting the optical ring image. Based on these values, according to equation(3), the measurement range of the sensor is about 1mm and the sensitivity is about $4 \mu\text{m}/\text{pixel}$.

The elliptic beam radiated from the laser diode(5mW, $\lambda = 780\text{nm}$) is modified to the parallel beam by a microlens L4 and it is illuminated on the target surface passing through the center of the object lens L1. The reflected light forms an optical ring in shape on the diffuse glass plate by passing through the ring slit and the converging lens L2. It is possible to reduce the speckle noise existing in the optical ring image due to average effect by rotating the diffuse glass plate at high speed. The diffused light through the rotating plate forms an average optical ring image magnified by lens L3 on the CCD areasensor. The intensity distribution of the optical ring image detected by CCD areasensor is stored in the image processor as 512×480 data points with 8 bit resolution. These data are dealt with by a computer(PC9801) with auxiliary memory.

The circumference of the detected optical ring image is divided into 24 equal parts at angle of $\theta = 15^\circ$. We search for a maximum intensity $I_{\max}$ and then the intensity I_t of 50% down from the value of $I_{\max}$ which means threshold. In the intensity distribution area to be larger than the the slice level(I_t), the weighted average value r_g is calculated in order to estimate the most correct peak position of the intensity distribution. The ring radius R is determined from the regression circle estimated by applying the least squares method to the 24 peak positions.

4. Sensor Properties

Figure 3 shows a typical optical ring image detected at the steeply inclined surface of 70 degrees. The resultant 3-D intensity distribution and the estimated regression circle are graphically shown. Even if the work surface is inclined at steep angle of 70 degrees, the optical ring image can be detected, as a result, the regression circle can be estimated correctly.

Figure 4 shows relationship between the set displacement and the estimated displacement which means radius of the optical ring image for both inclination angles of 0 to 70 degrees($\circ$) and 80 degrees($\bullet$) for plaster surface. The measurement point for the case of 0 to 70 degrees are almost the same values. The relationship can be expressed in a line and coincides with the analyzed result derived theoretically in chapter 2. For more steeply inclined surface of 80 degrees, the measuring error is yield. Figure 5 shows the measuring error of the displacement shown in Fig.4. The measurement for the inclined surface up to 70 degrees is considered to be as same as in the case of a flat surface. When the displacement is measured at more steep inclination angle of 80 degrees, the measuring error increases obviously compared with that at 70 degrees, however, it is found to be within an error range of $\pm 10\mu\text{m}$.

In order to check the feasibility of our proposed measuring method, the prototype of the optical ring image sensor is equipped with an NC machine tool called model master and the measurements are carried out for a steel ball surface. Fig.6 shows the 3-D profile of a steel ball surface and the resultant 3 different profiles. The radius of the regression sphere(7.934mm) is approximately equal to the value(7.938mm) obtained by 3-D

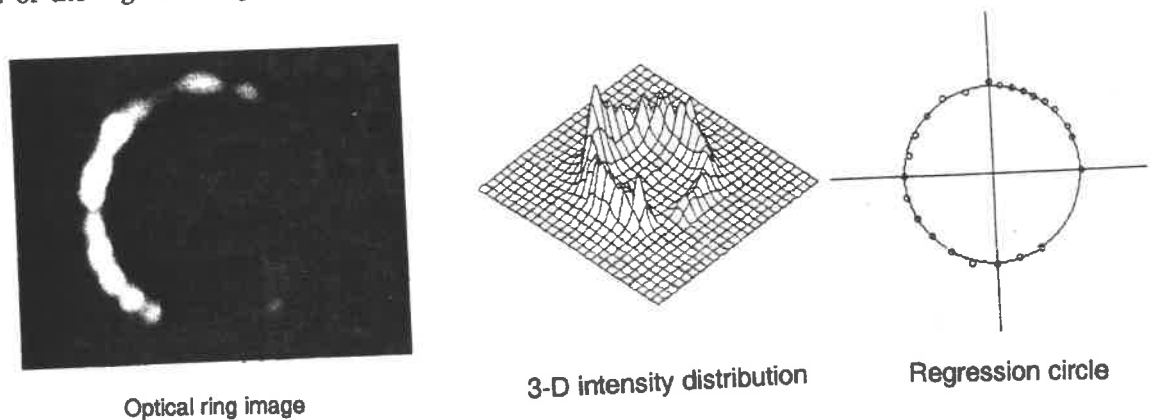


Fig.3 Optical ring image and regression circle for 70° steep inclination angle

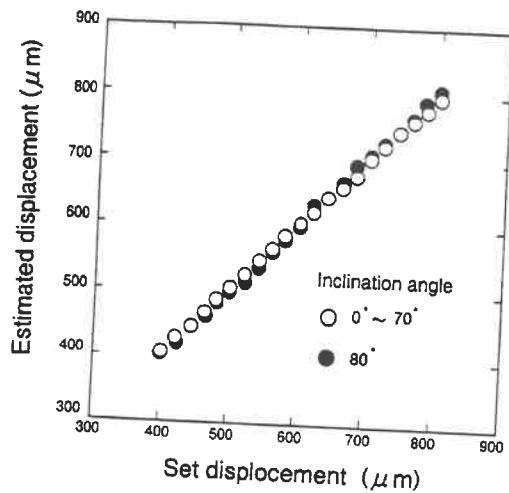


Fig. 4 Relationship between displacement and radius of the optical ring image for inclination angle of 0~80 degrees

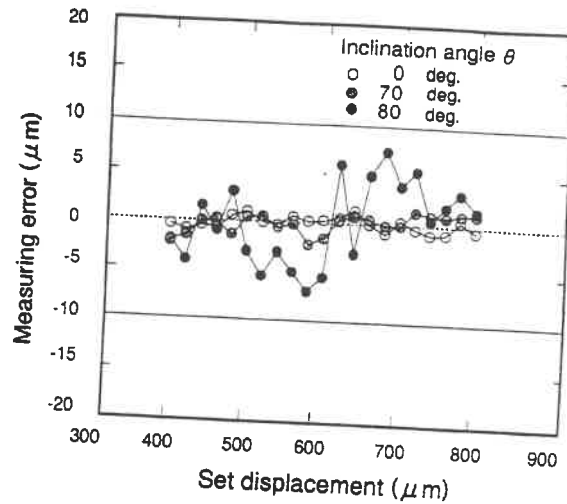


Fig. 5 Measuring error for inclination angle of 0.70 and 80 degrees

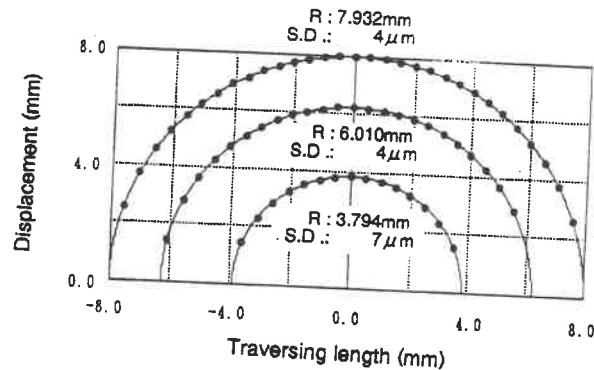
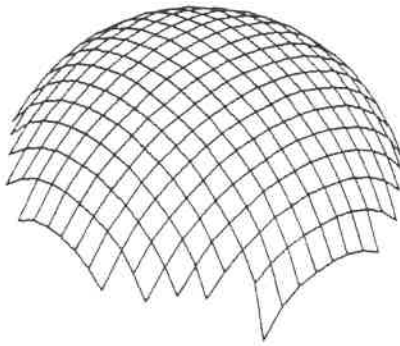


Fig. 6 3-D shape profile and 3 different radius profiles of a steel ball ($0.3\mu\text{mRa}$)

coordinate measuring machine. The standard deviation is not so large as to be $7\mu\text{m}$ which corresponds to the maximum measuring error of about $\pm 20\mu\text{m}$ although the measuring error has a tendency to become larger as the radius decreases. Accordingly, the optical ring image sensor makes it possible to measure even a nearly specular reflection surface up to about ± 70 degrees in inclination angle.

5. Conclusions

The results obtained are summarized as follows:

1. The radius of optical ring image sensor is linearly proportional to the displacement of the target surface in optical axis. This results coincides with the theoretical analysis.
2. The radius of optical ring image is accurately determined from the regression circle estimated by applying the least squares method to 24 peak positions obtained by weighted average method.
3. The speckle noise is eliminated from the optical ring image by passing the reflected laser light through a diffuse glass plate rotating at high speed.
4. This sensor makes it possible to measure the displacement within an accuracy of $\pm 10\mu\text{m}$ for the inclined diffuse reflection surface up to ± 80 degrees and also $\pm 20\mu\text{m}$ even for a nearly specular reflection surface inclined up to ± 70 degrees.

Inspection of the Diamond-Turned Surfaces Used for Mounting an Array of Eight X-ray Reflection Gratings

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Key Words: diamond-turning machine, dimensional metrology, X-ray Multi-Mirror project (XMM), snap-together optics, Precision Engineering Research Lathe (PERL)

Introduction

This paper describes the use of a T-base diamond-turning machine as a measuring machine for inspecting the positional accuracy of the diamond-turned surfaces of four attachment rails—parts that resemble precision step gauges. The attachment rails provide the precision mounting surfaces for a prototype array of eight X-ray reflection gratings for the European Space Agency's (ESA) X-ray Multi-Mirror project (XMM). Each rail is 4.5" long with a cross-section of less than 0.1 in², and has eight protruding bosses spaced approximately 0.5" apart (Figure 1). A diamond-turned feature on each boss provides a mounting surface for one of the four corners of a grating. These surfaces are 0.018" high by 0.1" wide, and have a 12" cylindrical radius with an axis parallel to the boss protrusion (Figure 2). Together, the four rails provide eight sets of four co-planar points for mounting the gratings (Figure 3). Note that the gratings are not parallel to each other; they sweep through a 12 mrad angle from the first to eighth grating. To accommodate this fanned array, the normal directions (denoted by arrows in Figure 1) of the mounting surfaces on the bosses, at the rail centerline, also sweep through a 12 mrad angle from the first to eighth boss.

The optical performance requirements of the grating array required that the mounting points locate the corners of the gratings within $\pm 15 \mu\text{m}$ of their designed positions. By virtue of the accurate location of the

mounting surfaces with respect to each other, the rails allow the gratings to be arrayed with the spacing and orientation accuracy necessary in a snap-together optics fashion; adjustments after installing the gratings are not required.

The base material of the rails is Ni-Fe Alloy 42 (invar), which has the same coefficient of thermal expansion as the gratings and the support structure ($2.4 \mu\text{in}/\text{in-}^\circ\text{F}$). The bosses were plated with 0.005" of copper to allow single-point diamond-turning of the grating mounting surfaces with minimal tool wear^a.

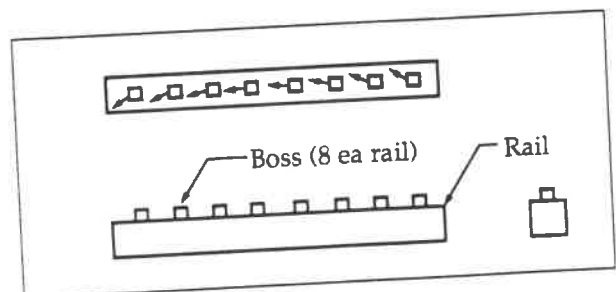


Figure 1. Rail with eight bosses.

^a See companion paper: *Fabrication of the Attachment Rails Used for Mounting an Array of Eight X-ray Reflection Gratings*, R.C. Montesanti, ASPE 1993 Annual Meeting Proceedings.

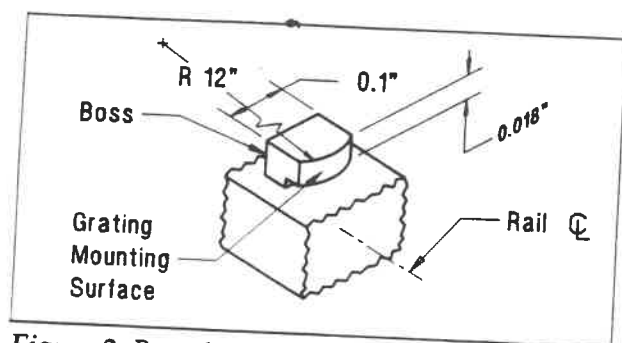


Figure 2. Boss detail.

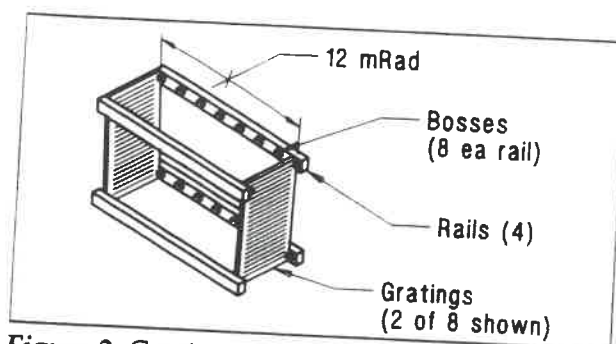


Figure 3. Gratings mounted to rails.

Inspection Set-up

The geometry of the eight cylindrical grating mounting surfaces on a rail was suitable for using a 2-axis measuring machine for inspecting the positional accuracy and form of the diamond-turned grating mounting surfaces. An error budget generated for the fabrication and inspection of the boss locations indicated that the error in the inspection process had to be less than $\pm 6.2 \mu\text{in}$. To achieve this precision, the Precision Engineering Research Lathe (PERL)—a 2-axis T-base diamond turning machine at LLNL—was used as a measuring machine. Figures 4 and 5 show a schematic of the set-up. A rail was mounted to the cross-slide so that the arcs along the grating mounting surfaces lay in the plane of motion of the PERL's cross-slide and spindle-slide (x- and z-axis slides, respectively). An air-bearing Linear Variable Differential Transformer (LVDT) probe was mounted to the z-axis slide so that it could measure displacements along the x-axis. The LVDT

amplifier was set for a $1 \mu\text{in}$ resolution, and its output was sent to a strip-chart recorder. The two slides were moved under computer control so that the LVDT probe tip traced along the designed arc of the grating mounting surface of each boss, one after the other, with a path error of less than $1 \mu\text{in}$. A location error of a boss with respect to the other bosses on the rail, or an error in its form would cause a recorded change in the deflection of the probe tip. If the 8 bosses on a rail had no location errors and were perfectly formed, then the LVDT probe deflections recorded on the strip-chart would be as shown in Figure 6. The rail passed inspection if the traces for all 8 bosses fell within the $\pm 13 \mu\text{in}$ tolerance band shown^b.

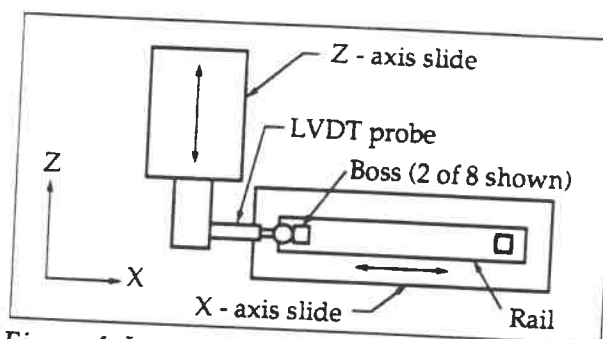


Figure 4. Inspection set-up.

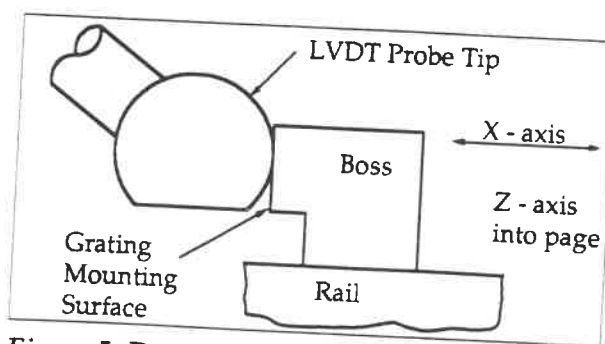


Figure 5. Detail of probe/boss contact.

^b Repeated measurements on one of the rails showed the measurement repeatability for the set-up to be within $\pm 2 \mu\text{in}$. The $\pm 13 \mu\text{in}$ tolerance band ensured that the boss locations would be within the required $\pm 15 \mu\text{in}$.

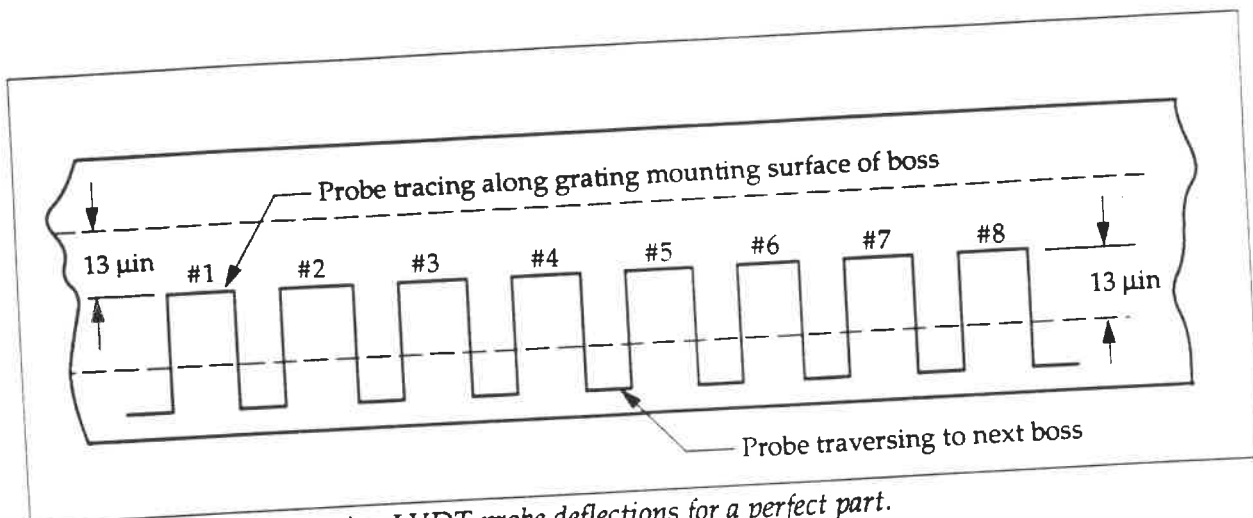


Figure 6. Strip-chart showing LVDT probe deflections for a perfect part.

Calibrating the PERL Machine

The PERL machine uses laser interferometers operating in ambient air for displacement measurements of the x- and z-axis slides. The wavelength of the light used varies with changes in the temperature, relative humidity, and pressure of the air. An enclosure surrounding the PERL machine maintained the temperature of the air to within $\pm 0.1^\circ\text{F}$ of a set-point temperature (nominally 68.0°F), and the relative humidity of the air remained constant to within $\pm 10\%$. The combined measurement errors due to those two factors were less than $\pm 1\text{ }\mu\text{in}$. Although the air pressure remained constant (to within 0.05 in-Hg) during the measurement of a rail, its absolute value changed significantly from rail to rail. Wavelength compensation was entered into the PERL machine's software to account for those changes.

The displacement measurement system was calibrated against a set of certified Tungsten-Carbide^c Grade II gauge blocks. Since the length standard and the parts to be measured have a similar coefficient of thermal expansion, a small ($\approx 1^\circ\text{F}$) error in

the absolute temperature at the time of calibration was not critical. Because the grating array performance is unaffected by a small ($\approx 10\text{ }\mu\text{in/in}$) consistent scaling error in all four rails, the calculated uncertainty of $\pm 3\text{ }\mu\text{in/in}$ in the absolute length calibration was acceptable. To insure consistent measurement results between all four rails, the measured boss spacings were scaled to the same reference temperature of 68.0°F .

A static drift test was performed to ensure that the PERL machine's laser interferometers, positioning servos, and thermal control were adequate for achieving a positioning accuracy on the order of $5\text{ }\mu\text{in}$. Both the x- and z-axis slides were activated and commanded to maintain a static position. Since the positioning accuracy of the x-axis slide was critical for measuring the location of the grating mounting surfaces, an LVDT was set up to measure displacements in the x-direction of the x-axis slide relative to the z-axis slide. Drift of the x-axis slide over 30 minutes was less than $2\text{ }\mu\text{in}$, which was acceptable.

^c Tungsten-Carbide has a coefficient of thermal expansion that is almost identical to that of the Ni-Fe Alloy 42 ($2.4\text{ }\mu\text{in/in-}^\circ\text{F}$).

Inspection results

Table 1 shows the measured location errors of the grating mounting surfaces on the four rails. The errors for all four rails were within $\pm 11 \mu\text{in}$. Repeated measurements on one of the rails indicated that the inspection process was repeatable to within $\pm 2.0 \mu\text{in}$. Therefore, all four rails met the required $\pm 15 \mu\text{in}$ tolerance for the parts. The error span for the first four rails indicated that a progressive degradation in the diamond-turning process was occurring. We suspected that the diamond tool was beginning to exhibit appreciable wear, so it was replaced with a new tool before machining the fourth rail. The $\pm 3.1 \mu\text{in}$ span on the boss locations for the fourth rail verified those suspicions. The location of the grating mounting surfaces on one of the rails was measured at two different heights above the long axis of the rail (in-feed direction of the fly-cutter). This comparison showed that these surfaces were parallel to each other to within $5 \mu\text{in}$ in that direction, which was acceptable. Because the nominal embedment of the bosses into the grating is $20 \mu\text{in}$ for the designed 1 lb clamping load^d, observed form deviations from a smooth arc, with amplitudes of $\approx 2 \mu\text{in}$ or less, would not adversely affect the mounted locations of the gratings.

At the time of this writing, preliminary optical tests using a Fizeau interferometer

^d The grating surface is actually a replica — $500\text{\AA}$ of gold on a $40 - 60 \mu\text{m}$ thick layer of oven-cured epoxy — of a master grating.

	Boss 1	Boss 2	Boss 3	Boss 4	Boss 5	Boss 6	Boss 7	Boss 8	Span
Rail 1	+ 4.0	+ 4.8	+ 1.2	- 2.5	- 2.7	- 4.8	- 1.0	- 1.2	± 4.8
Rail 2	- 4.9	- 7.8	- 7.1	- 3.0	+ 5.7	+ 7.8	+ 1.9	- 2.9	± 7.8
Rail 3	- 11.0	- 6.7	- 3.9	+ 4.9	+ 11.0	- 2.5	- 2.6	- 1.3	± 11.0
Rail 4	- 0.6	+ 3.1	- 0.2	- 1.5	- 1.7	- 1.0	+ 0.7	- 3.1	± 3.1

Table 1. Location errors of the grating mounting surfaces (μin).

This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under contract No. W-7405-Eng-48.

and a precision rotary table indicated that the as-mounted orientations of the eight gratings mounted against the bosses of the four rails (EOBB array) were better than what was predicted for the case where the corners of each grating had a $\pm 15 \mu\text{in}$ location error. The final proof of the accuracy of the boss locations will occur in October 1993, when the X-ray performance of the EOBB array will be tested in Europe.

Conclusion

A T-base diamond-turning machine was successfully used as a measuring machine for inspecting the positional accuracy of the diamond-turned surfaces of the four attachment rails used for mounting eight X-ray reflection gratings into an array. The inspection process revealed that the eight grating mounting surfaces on each rail were machined to a positional accuracy of $\pm 11 \mu\text{in}$ or better, meeting the required $\pm 15 \mu\text{in}$ tolerance for the parts.

Acknowledgments

Dick Grayson of LLNL calibrated the PERL machine and performed the rail inspections with it. Charlie Giles of LLNL wrote and debugged the computer programs used to inspect the rails with the PERL machine. Special appreciation goes to Pete Davis, Todd Decker, and Bob Donaldson, all of LLNL.

ULTRA-COMPACT OPTICAL PICKUP MECHANISM

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1. Introduction

The MD (Mini Disc) has been absorbing public attention as a next generation audio product, because it can provide 74 minutes of high quality digital sound recording and playback just like a Compact Disc (CD) in spite of its smaller size.¹ Moreover, since the MD can hold one hundred and several tens of megabytes digital data, it is expected to be a compact data storage device for personal computers.

Last year, we announced the development of a thin optical pickup, whose thickness was 8.7mm. In addition, a pocket-sized player for MD playback and an autochanger player for car audio were also developed with this optical head.² However, further miniaturization of the optical pickup was required for assembling more compact MD products. To meet this requirement, we have redesigned the optical path layout and the mechanical structure. As a result, we have succeeded in realizing thinner optical pickup this time. The thickness of only 7.9mm is realized while the high cost-performance is achieved.

2. Optical pickup

The external appearance of the optical pickup is shown in Figure 1 and the main specifications are listed in Table 1. The external dimensions are 45 x 42 x 7.9mm and weight is 15.5g. The three main points in realizing such

an ultra-thin, lightweight optical pickup are as follows:

1. Simplification and reduction of thickness of the optical system
 2. Miniaturization of the pickup mechanism
 3. Miniaturization and improvement of performance of the objective lens actuator
- These points will be explained in the following sections.

3. Optical design

To achieve a reduction in thickness, the optical system utilizes a compact aspheric molded-glass lens to reduce the optical beam diameter to $\phi 2.7\text{mm}$. Further, by utilizing a compact high-output semiconductor laser as the light source, it became possible to reduce the overall size of the optical system. In addition, a distinctive feature of the molded-glass lens is its superior resistance to environmental conditions.

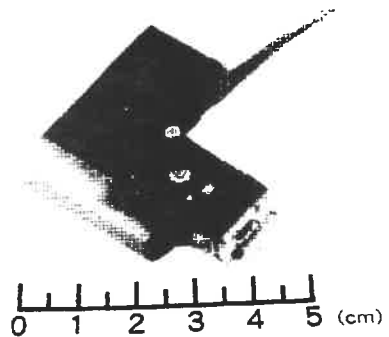


Fig. 1 Appearance of the optical pickup

Table 1 Specifications of the optical pickup

Item		Specifications
Type		MLP-S1
Laser Diode	Type	ML60111R
	Output	35mW
	Wavelength	785nm
Objective Lens	NA	0.45
	WD	1.4mm
Output Power		5mW(min.)
Direction of polarization		E-vector perpendicular to pregroove
Error Detection Method	Focusing	Astigmatic
	Tracking	3-Beam
HF Module		550MHz
Lens Actuator		Sliding & Rotary Type
Dimensions		45(W)x42(D)x7.9(H)mm
Weight		15.5g

A high-reliability and low-cost sensor system is adopted, utilizing an astigmatic focusing and a 3-beam tracking method.

In order to reduce the number of components as well as simplify adjustment, we designed the optical head to carry a 3 beam wollaston prism and multielement photo detector composed of 8 elements for signal and focusing/tracking error detection.

Further, a tiny high-frequency module was developed so that laser noise during playback can be suppressed to a non-problem level by high-frequency superposition current at approximately 550MHz.

On the upper surface of the optical pickup body, a headamp IC is attached, which enables a sufficient carrier to noise ratios even for the minute magneto-optical signals from recorded discs.

Figure 2 shows the arrangement of the optical systems.

4. Pickup mechanism

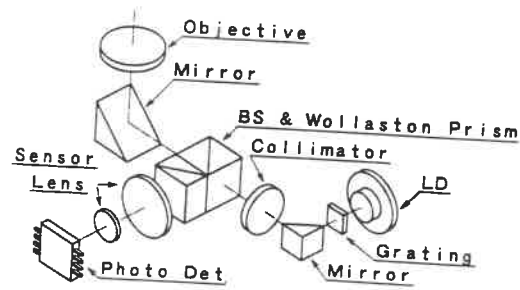


Fig. 2 Optical path diagram

In addition to reducing the size and thickness of the optical pickup base, the shape is selected to also provide extremely wide freedom in the structural design.

In order to make the height of the optical pickup as thin as possible, the position of the central optical axis was decided based on a comprehensive evaluation of not only the optical components but of all components, such as the optical detector, objective lens actuator components, etc. For this reason, a structure with fixed optical components installed on the underside of the pickup base is used. A similar method is used for installing the high-frequency module.

The width of the optical pickup is reduced by bending the converging optical system and an external shape in which the optical pickup stays away from the disc-cartridge positioning pin even while moving from inner tracks to outer tracks is realized.

For the disc motor, by planning the arrangement of the optical components and reducing the width of the objective lens actuator to stay away from the disc motor section, it became possible to use round motors up to ϕ 16mm, providing greater freedom in the selection of the motor.

Further, the optical pickup support section utilizes a ϕ 2mm shaft with a shaft pitch of 47mm. A sliding pivot is conceived for the transmission mechanism section, but a removable structure is selected. Because of this, even when a high-speed transmission mechanism such as a linear motor is utilized,

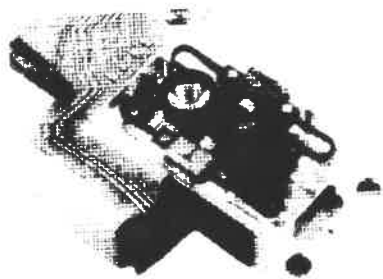


Fig. 3 Appearance of the objective lens actuator

Table 2 Specifications of the objective lens actuator

Item	Specifications	
	Focusing	Tracking
Moving Distance(min.)	$\pm 0.45\text{mm}$	$\pm 0.5\text{mm}$
Acceleration	27G/A	25G/A
Parasitic resonance freq.	Over 30KHz	Over 30KHz
Resonance freq.	19Hz	38Hz
Coil DC Resistance	5 Ω	5 Ω

the main unit of the optical pickup can easily be used as is, without changing this section.

5. Objective lens actuator

The objective lens actuator uses this company's exclusive sliding and rotary type to realize high performance while greatly reducing size. The advantage of this system is that its positional deviation has been reduced to a minimum when performing high-speed access with the optical pickup, such as for data storage devices.³ Figure 3 shows the external appearance of the objective lens actuator, and Table 2 lists the main specifications.

To reduce the size of the objective lens actuator, it is necessary to miniaturize both the movable lens holder section and the fixed magnetic circuits. On the other hand, the desired control bandwidth of the servo system



Fig. 4 Bending vibration mode(33.8KHz)

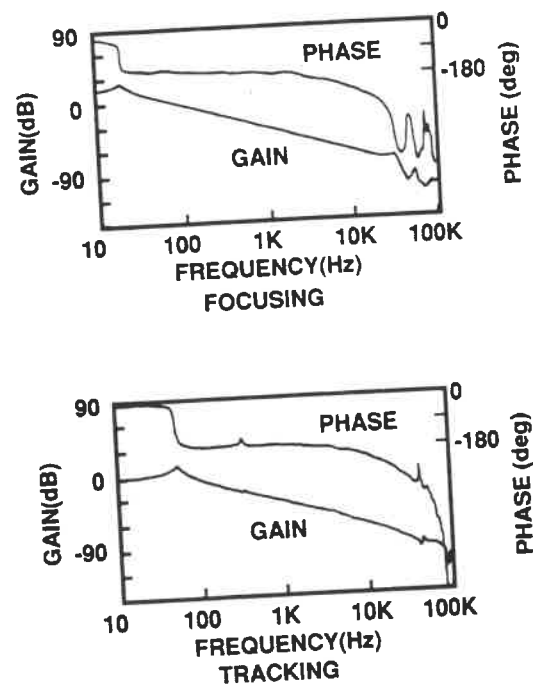


Fig. 5 Frequency characteristic of the objective lens actuator

is at least 1KHz, but it is necessary to eliminate parasitic resonance accompanying the increase in amplitude and the large phase shift in the frequency region of approximately ten times the control bandwidth. Further, to reduce power consumption, it is necessary to make the driving force larger. To satisfy these inconsistent aims, we devised the following mechanism.

To reduce the size of the movable lens holder section, the shaft diameter is reduced to $\phi 1.5\text{mm}$, and by using an aspheric molded-glass lens as the objective lens, the eccentricity of the objective lens in relation to the shaft is reduced.

Epoxy resin with a high specific rigidity is used as the material for the lens holder. To determine the most suitable shape for the lens holder, detailed structural analysis was performed utilizing the finite element method. This resulted in a parasitic resonance frequency of more than 30KHz and a control bandwidth with a high value of more than one digit despite a minimum thickness of only 1.8mm. The result of solid element analysis for bending vibration mode is shown in Figure 4. The weight of the movable section including the objective lens and drive coil is only 0.45g. Figure 5 shows the measured results for the frequency characteristics in the focusing direction and tracking direction. Suitable characteristics with no parasitic resonance to over 30KHz are obtained in both focusing and tracking directions.

To make the drive system highly efficient and reduce mutual interference, drives for the focusing direction and tracking direction are separated. To reduce the width of the objective lens actuator, a construction in which the tracking coil is enclosed in the lens holder is used. As a result of the light weight of the lens holder and achieving the most suitable magnetic circuit structure in both directions according to detailed three-dimensional analysis of magnetic fields utilizing the finite element method, a high acceleration per unit electrical current of more than 25G/A (G: Acceleration due to gravity) is achieved in both directions using a rare-earth magnet of 0.02cc in the focusing and 0.03cc in the tracking. Further, to avoid placing the magnetic circuits directly under the objective lens, a reflective mirror is installed, preventing an increase in the height of the objective lens actuator.

When using the sliding and rotary type actuator, there is a possibility of misalignment due to play and friction between the shaft and the sleeve causing a deterioration of the motion or control characteristics.⁴ The clearance between the shaft and the sleeve is

approximately 10 μ m. The friction coefficient between the two is low, less than approximately 0.3. It was verified that the influence of friction could be totally neglected. As a result, an actuator volume of approximately 1/8 of that of our company's CD optical pickup was achieved and a low-power-consumption, high-performance objective lens actuator which is not only ultra-thin but is also suitable for the narrow cartridge window of MD was developed.

6. Conclusion

Based on the above studies, we have developed a recording/playback optical pickup for MD which is not only ultra-compact and thin but also offers high performance at low cost. This optical pickup has a total volume of approximately 1/3 and an objective lens actuator size of approximately 1/8 of those of our company's CD optical pickup, and its weight is only 15.5g.

By utilizing this optical pickup, it will be possible to realize a disc-cartridge-size deck mechanism.

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Development of a Virtual Numerical CMM in Pursuit of Accuracy Assessment for CMMs

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1.Introduction

Accuracy assessment such as verification, calibration, and compensation are becoming more essential elements for manufacture and maintenance of CMMs aiming high performance and accuracy. In pursuit of accuracy evaluation for most commercially available CMMs, a virtual numerical CMM has been developed in this research, and the basic features are as follows:(1) The virtual CMM operating on a micro computer environment accepts the information on parametric errors along machine axes, e.g. 21 parametric error components and thermal errors for 3 axis CMMs.(2)A set of complete volumetric error equations are constructed so that the volumetric accuracy can be assessed within a working volume of the machines. Various kinematic configurations are considered, thus it can be applied to most types of commercially available CMMs. The effect of measurement offsets of parametric errors on the volumetric error also has been considered, and appropriate transformation rules are implemented with respect to the origin of the working volume. (3) The calculated volumetric accuracy form the volumetric error map, and is represented in a form of 3 dimensional grid patterns, in which actual grids as well as nominal grids are displayed in 3D pattern.(4) Various measurement simulation modules are implemented ; such as step gauge, ring gauge, sphere, and cylinder, etc., which are to assess the influence of the parametric errors on the practical measurement tasks commonly employed in the practical usage.

2.Calculation of volumetric error map for 3 axis CMMs

The 3 dimensional volumetric error behaviour has been focused in the sense of error verification, calibration, and compensation, e.g.[4]. The calculation of volumetric errors, however, needs the detail kinematic chain of machine structure, coordinate axes configuration, and measurement location, hence it was very much dependent on the specific machines in current use. Thus a generally applicable volumetric error calculating algorithm has been developed, which can input parametric errors from most of commercial CMMs, then calculate the volumetric error map for the machines.

As various types of CMMs are in current use, the developed module aims to cover the various types and configuration of commercial CMMs. Many national standards ([1],[2],[3]) describe several models. In this research 10 kinematically different models of 5 types are classified and implemented depending on the machine kinematic configuration and coordinate axes arrangements: Moving Bridge Type (2 models : MB1, MB2) , Fixed Bridge Type (2 models:FB1,FB2), Moving Horizontal Arm Type (2 models:MH1,MH2), Fixed Horizontal Arm Type (2 models:FH1,FH2), and Column Type (2 models:CT1,CT2). For the 10 kinematically different models, a set of complete volumetric error equations are derived and implemented, and thus proper model can be chosen for the specific machine by the user. Systematic error and random error which are depending on the reproducibility of datum, thus have to be considered for the virtual CMM work. The developed module

accepts the systematic error and random error at the stage of parametric error input; where both errors can be entered as numerical input or error band input.

Error transformation rule

In general, measured parametric errors such as positional, straightness, and squareness errors are varying with the measurement location, which is the measurement offset. For example, two X positional errors at two different measurement locations may not give identical values, because some angular errors may be associated with the measurement offsets, which is the distance between the two measurement locations. It is also unavoidable to allow some extent of measurement offsets due to physical and mechanical limitations such as mechanical and optical obstacles. In this research, a set of empirical transformation rules have been derived for the 10 kinematically different models, using the volumetric error equations. It is of metrological importance to note that the positional error, straightness error, and squareness errors are generally influenced by the associated angular errors having measurement offsets. For example, (X_o, Y_o, Z_o) be the reference point or origin of working volume, where all parametric errors have to be transformed; (X, Y, Z) be the practical measurement location within the working volume. In case of moving bridge type CMM, the X positional error, Y positional error, X straightness error along Y axis, and the XY squareness error can be transformed as follows:

$$\delta^*x(x) = \delta x(x) - (y - y_o)E_y(x)$$

$$\delta^*y(y) = \delta y(y) + (z - z_o)E_x(y) - (x - x_o)E_z(y)$$

$$\delta^*x(y) = \delta x(y) - (z - z_o)E_y(y)$$

$$\alpha^* = \alpha + E_z(y) - (z - z_o)E_x(x)/(x - x_o)$$

where $\delta x(x), \delta y(y), \delta x(y)$ are X positional, Y positional, and straightness errors, and $E_y(x), E_x(y), E_z(y), E_y(y), E_z(y)$, and $E_x(x)$ are X pitch, Y pitch, Y roll, Y yaw, and X roll errors, respectively, and α is the squareness error between the X, Y axis ; while the "*" indicates the transformed parametric errors.

Volumetric error map

The input parametric errors are processed to give the volumetric error for a measurement volume considered, after possible transformation with respect to the origin point. The systematic part can be calculated straightforward from the derived error equations. For example, the moving bridge type CMMs have the volumetric error equation of X axis as follows([4],[5])

$$\Delta X = \delta x(x) + \delta x(y) + \delta x(z) + z (-\beta_1 + E_y(x) + E_y(y)) - Y_p (E_z(x) + E_z(y) + E_z(z)) + Z_p (E_y(x) + E_y(y) + E_y(z)) + K_x T_x$$

where b_1 is the squareness error between the X and Z axis, X_p, Y_p, Z_p are the coordinates of the measurement probe, and K_x, T_x are the thermal expansion coefficient and the temperature increase of X axis from a reference temperature. The random part can be evaluated using the error propagation rules of quadrature formula. The random part of the volumetric error is

$$R\Delta X_i = \sqrt{\sum ((\partial \Delta X_i / \partial E_j) \Delta RE_j)^2}$$

where ΔX_i ($i=1,2,3$) is systematic part of volumetric error, E_j ($j=1,2,..,21$) is systematic part of the j th parametric error, ΔRE_j ($j=1,2,..,21$) is random part of the j th parametric error.

The calculated volumetric error can now be displayed in a form of 3 dimensional

map, where the volumetric error map is plotted in 3D grid pattern, and original nominal grids being plotted in different colors within the specific working volume.

3. Simulation of geometric features

In conjunction with the calculation of volumetric error map, it is also of metrological importance to investigate the influences of the machine errors on some specific measurement tasks such as step gauge, ring gauge, and sphere measurement. Step gauge and ring gauge are simple, useful equipments for assessment of machine accuracy. As the error informations are known, it is possible to simulate the step gauge measurement (which is length measurement error) and ring gauge measurement. When the length measurement errors are evaluated at various length intervals, it is possible to plot a graph of error vs. nominal length, displaying the length measurement error at every length interval. Many national standards such as BS and VDI/VDE recommend some representative measuring lines for the length measurement error using step gauge or laser interferometer, and the investigation on the representativeness of the selected measuring lines is of metrological interest. Hence a specific module has been developed in this research to assess the length measurement errors along the representative measuring lines, all possible measuring lines, and randomly chosen measuring lines; and the comparisons are made.

Similarly, ring gauge measurement simulation has been performed, showing the influence of parametric errors on the ring measurement tasks. The ring gauge, cylinder, sphere measurements are also simulated, thus the influence of the parametric errors on the specific measurement tasks can be shown.

4. Practical application and conclusion

The developed module has been applied to practical machines in use, in order to assess the effectiveness and practicability. A CMM of moving bridge type is chosen, whose working volume is 900mmX1150mmX400mm, and 17 parametric errors were measured using laser interferometer, precision level, and mechanical artefacts. Fig1 shows the 3 dimensional display of the calculated volumetric error map, giving length uncertainty values of -24.0 μ m, 21.3 μ m, 25.9 μ m, and -35.9 μ m along the 4 space diagonals. The step gauge measurement simulation has been performed, and it is noteworthy to mention that the length uncertainty values along some representative measuring lines are much less than those along all possible measuring lines, and a number of randomly chosen measuring lines can give very close values for all measuring lines. Fig.2 shows the step gauge measurement along the 4 space diagonals. Fig.3 shows the ring gauge simulation of 150mm radius, 200 sampling points are considered along the circumference. Fig.4,5 show the cylinder and sphere gauge measurement simulation.

Conclusions of the research is as follows:

- (1) A computer module for volumetric error calculation is developed, which is applicable to most types of commercial CMMs in use.
- (2) A set of general transformation rules are developed enabling the measured parametric errors to be transformed w.r.t the machine reference.
- (3) Measurement simulation is performed for step, ring, and sphere gauge. The influences of the parametric error on the specific measurement tasks are investigated.
- (4) Several cases of length measurement along measuring lines are assessed, and the representativeness of length measuring lines is investigated. Length measurement along some representative measuring lines gives much smaller values

than along all possible measuring lines within a working space.

Acknowledgements

The authors are grateful to Dr.G.Peggs of NPL,UK for his kind assistance and advices on this research project.

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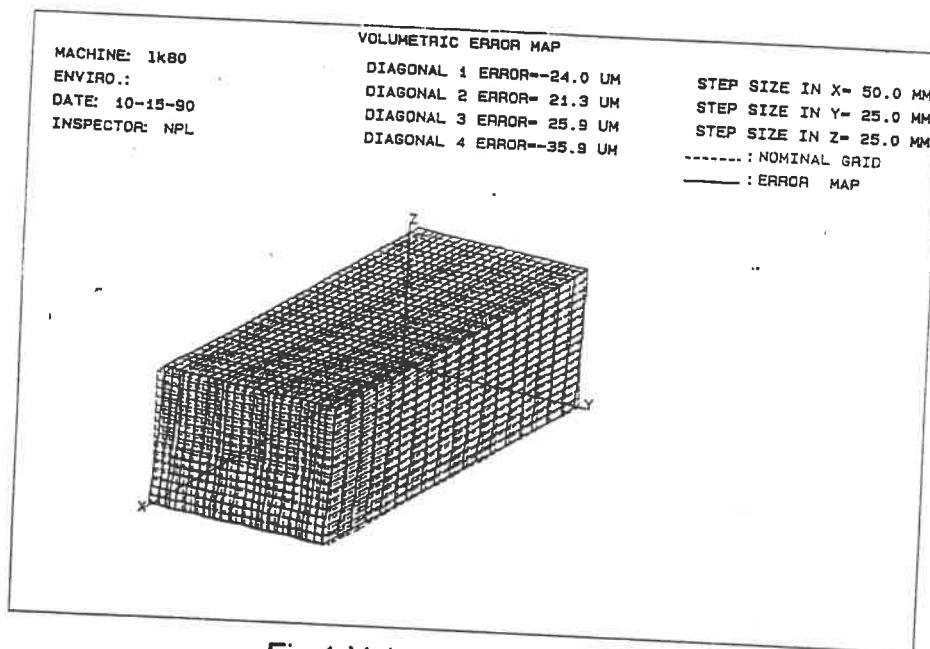


Fig.1 Volumetric Error Map

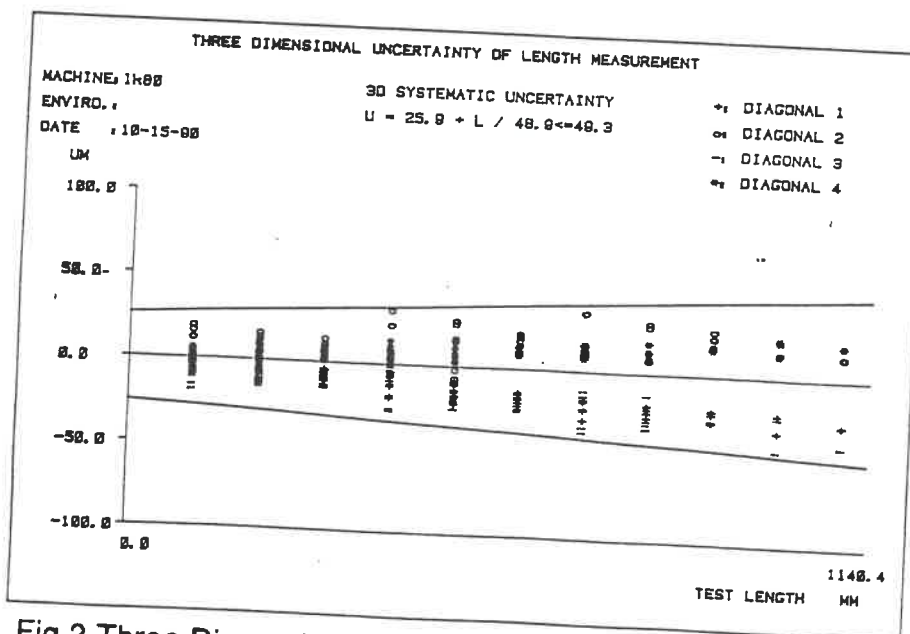


Fig.2 Three Dimensional Uncertainty of Length Measurement

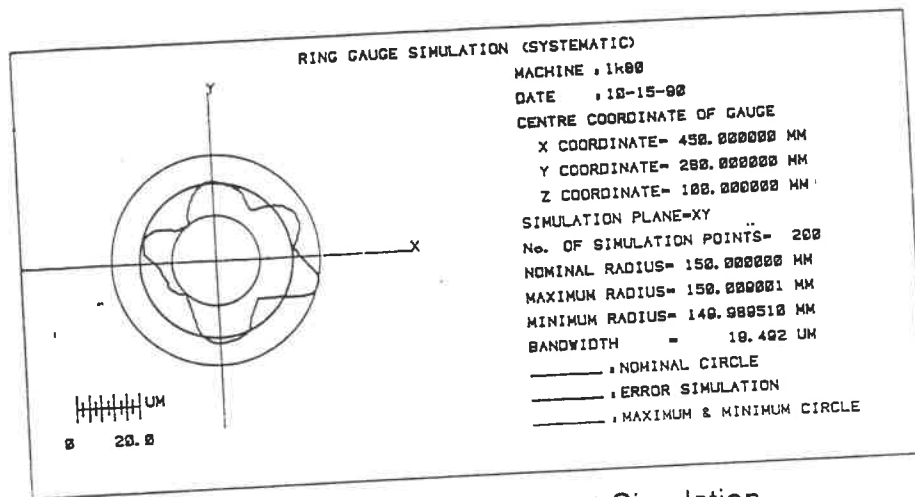


Fig.3 Ring Gauge Measurement Simulation

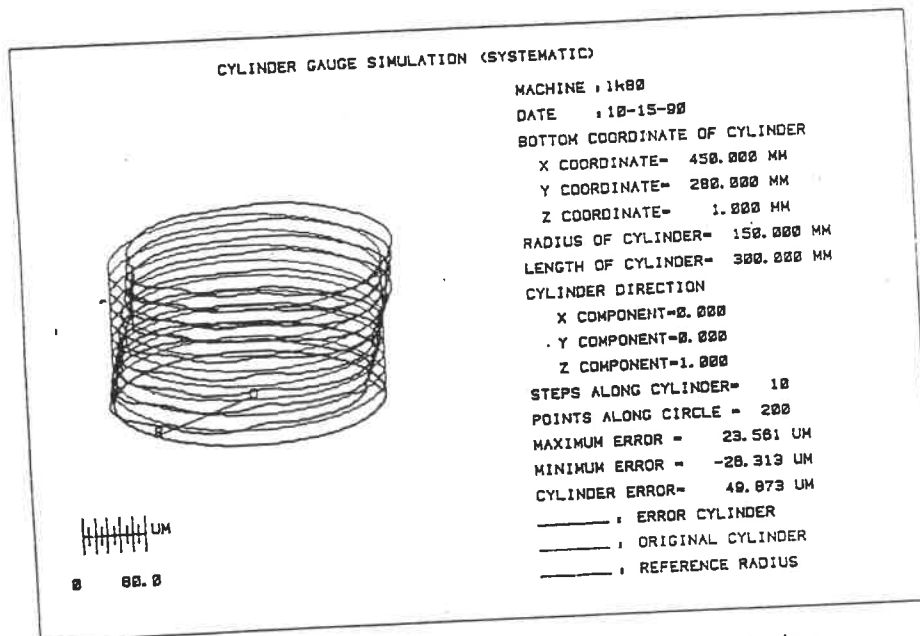


Fig.4 Cylinder Gauge Measurement Simulation

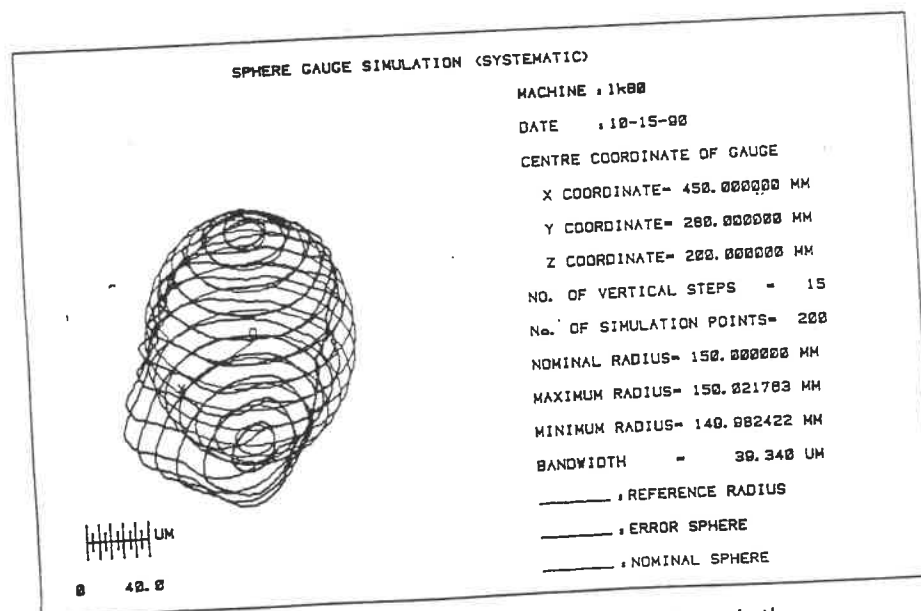


Fig.5 Sphere Gauge Measurement Simulation

A CMM INTERIM TESTING ARTIFACT DEVELOPED BY NIST

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Introduction and Background: The Coordinate Measuring Machine (CMM) interim testing program at the National Institute of Standards and Technology (NIST) originated from an Air Force request to develop a comprehensive CMM testing program which can be implemented on a regular basis. The need for a regular testing program, now known as interim testing, was identified after a significant number of CMMs at one Air Force base failed the annual recertification procedure. This demonstrated that out of calibration CMMs had been in operation for an unknown amount of time, perhaps as long as one year, which may have resulted in the rejection of good parts or the acceptance of bad ones. This program is a cooperative effort between NIST, DoD, DoE, and private industry.

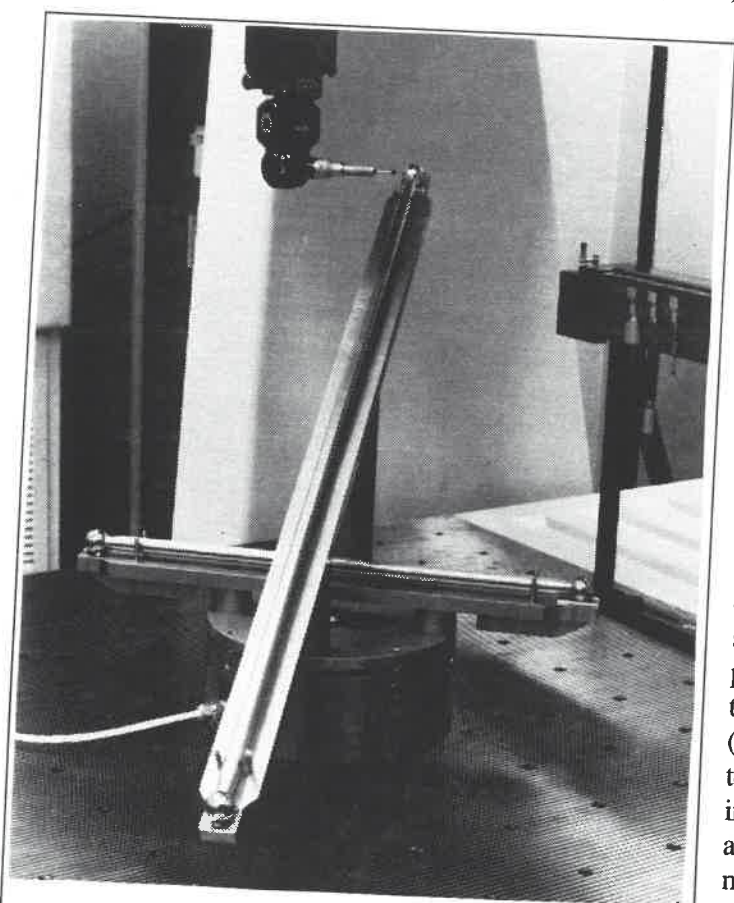


Figure 1: The NIST CMM interim testing artifact shown using two calibrated ball bars.

The goal of CMM interim testing is to identify and rapidly remove from service defective CMMs. The frequent application of interim testing will increase confidence in CMM performance between CMM calibrations. Interim testing is not a substitute or replacement for CMM calibration, rather it checks the validity of the calibration by detecting common CMM performance failures. As interim testing is the responsibility of the CMM user, special attention was directed at identifying what criteria CMM users felt were essential to a successful interim testing program. In particular, we have specifically designed this artifact to: (1) examine the entire CMM measurement system, including subsystems such as probes, indexable probe heads, temperature compensation equipment etc.; (2) minimize the time required for the testing routine; (3) be relatively inexpensive; (4) be lightweight; (5) be adaptable to different CMM sizes and machine designs.

Artifact Design: The NIST interim testing artifact, shown in Figure 1, is a pseudo three dimensional (3D) calibrated ball plate. Unlike an actual 3D ball plate, our artifact is lightweight, relatively inexpensive, easy to calibrate, and adaptable to different sizes and styles of CMMs. The artifact is composed of ball bars which are calibrated for ball roundness, ball diameter, and sphere center-to-center length. These ball bars are kinematically mounted on supporting arms and can rotate (in 45 degree increments) to sweep out most of the CMM work zone. The horizontal and inclined arms are interchangeable and are available in lengths from 300 mm to 1.5 m.

Central to our artifact design is the functional distinction between dimensional calibration and structural rigidity. Many conventional CMM artifacts such as ball plates utilize heavy structural components to insure the accuracy of the calibrated dimensions. The NIST interim testing artifact separates these functions through the use of kinematically mounted ball bars. Each calibrated ball bar is held kinematically by a support arm, with one ball of the ball bar held in a three point mount and the other ball held in a two point mount. This arrangement has several advantages. Ball bars are long thin artifacts which are subject to bending and vibration that may degrade the measurement result. Supporting the ball bar directly under each ball allows the ball bar to provide an accurate calibrated length while having the support arm provide the structural rigidity. Hence this is an efficient division of labor; the ball bar provides the dimensional accuracy requirements and the supporting arm supplies the structural requirements. It also means that the calibrated component, *i.e.* the ball bar, can be easily removed for recalibration without having to be concerned with the underlying supporting frame. The kinematic nature of the ball mounting means that the ball bar is uniquely located in the arm with no possibility of rocking or other unstable motion (probing forces are much less than the mounting force so the system is nearly kinematic when measured). The kinematic mounts also isolate the ball bar from any possible distorting forces. Hence the supporting arm could thermally expand, or otherwise distort and the ball bar will remain unaffected. Consequently, the entire supporting frame can be constructed from inexpensive materials such as aluminum (which has a high stiffness to weight ratio) without concern for dimensional accuracy or stability.

One of the significant drawbacks of using a traditional ball bar with computer controlled CMMs is that the CMM must be manually taught the location of the ball bar before it can be measured under computer control. Since this procedure must be carried out at each location this is a slow laborious process. The NIST interim testing artifact is designed to locate on one of the threaded inserts on the CMM table. This locating capability allows the artifact to be repeatably located within the "seek distance" of the CMM, typically a few millimeters. Hence, once a computer program is written to carry out the interim testing, this same program can be used to measure the artifact for all future tests without any manual probing points being required. Consequently, the interim test can be run very rapidly on DCC CMMs.

The NIST CMM interim testing artifact employs calibrated ball bar(s) which are calibrated for ball roundness, ball diameter, and sphere center-to-center length. The ball bar was selected as the calibrated component of the interim testing artifact for several reasons listed below.

- The highly spherical balls of the ball bar are accessible from many different probe approach directions which is useful in assessing probe performance and other short range CMM errors.
- The ball diameter provides a calibrated length for an explicit bi-directional length measurement.
- The center-to-center length provides a long calibrated length to assess CMM geometry errors.
- The geometry of the ball bar leads to a natural deconvolution of CMM errors (as described later).
- The length of the ball bar/supporting arm can easily be made comparable to that of the CMM axis which is desirable, as long artifacts are more sensitive to angular errors.
- High precision (0.15 μm) spheres are inexpensive, hence ball bar production costs are low and ball bars are easy and inexpensive to calibrate.
- The measurement of the ball bar does not have any alignment problems, *i.e.* cosine errors which can occur with artifacts such as gage blocks; in this sense the ball bar is self aligning.

The geometry of the ball bar leads to a natural deconvolution of CMM errors. When a sphere is measured by a CMM many points can be taken and then used in the sphere fit. The difference between

the best fit radius of the sphere and the "radius" from each of the measured points to the best fit sphere center can be found; we call these differences the radial deviations. These radial deviations are primarily a result of short range CMM errors such as repeatability or probe problems. The sphere diameter, which is determined using all of the measured points, averages over all the radial deviations and hence is largely independent of these effects. The diameter deviation is the difference between the best fit sphere diameter and the calibrated diameter; this checks the effective stylus diameter and hence the probe calibration artifact. The (fitted) sphere center location is independent of the effective stylus diameter, and largely (because of averaging) independent of the radial deviations. Consequently, the center-to-center distances between the balls are primarily a result of long range errors such as CMM geometry problems. Hence the geometry of the ball bar artifact naturally allows short range errors, probe calibration errors, and long range errors to be conveniently sorted out, as shown in Table 1.

Table 1 Separation of CMM errors using a calibrated ball bar

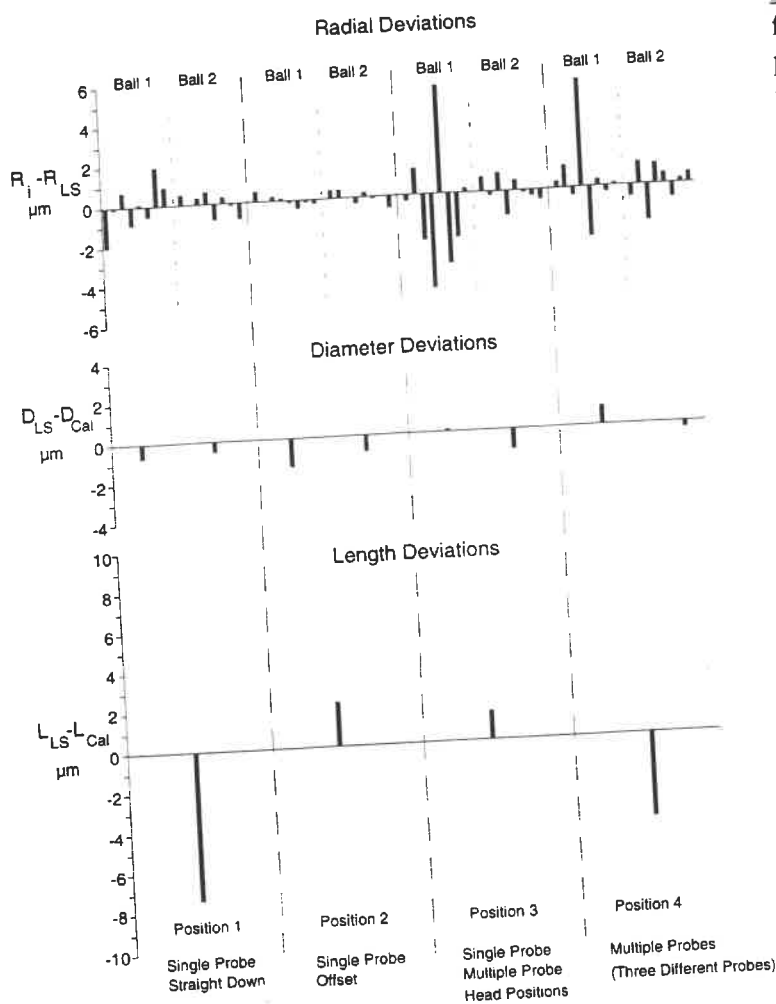
Radial Deviations	• CMM Repeatability • Probe Repeatability • Probe Lobing • Stylus Ball Form, <i>i.e.</i> out-of-roundness • Stylus Bending • CMM Dynamics (Vibrations) • Probe Head Repeatability • Probe Offset Vector Errors (multiple styli/probes) • Probe Changer Repeatability • Short Range Scale Errors
Ball Diameter Deviations	• Effective Stylus Diameter (probe calibration)
Center to Center Ball Bar Length Deviations	• CMM Geometry (Squareness, Pitch, Roll, Yaw, etc.) • CMM Scale Temperature Compensation • Part Temperature Compensation

Sample Testing Procedure:

A basic test for a three axis CMM with an indexable probe head and a probe changing rack involves using one ball bar in the 4 body diagonal positions of the CMM. In each position the user decides to take eight points on each ball of the ball bar. The user calculates radial deviations, *i.e.*, the differences between the best fit ball radius and the distance from the best fit sphere center to each

probing point. Similarly, the difference between the best fit sphere diameter and the calibrated diameter is recorded for each sphere. Finally, in each of the four ball bar positions the differences between the center-to-center lengths and the calibrated value are recorded. These results are shown in Figure 2.

In the first body diagonal position the user employs a single probe (oriented along the ram axis). In this position the radial deviations show the repeatability of the CMM and of the probe, and any probe lobing effects. The ball diameter measurements checks the probe calibration, *i.e.*, the effective stylus ball size, and the short range errors in the CMM scales. The bar (center-to-center) length measurement checks for long range (CMM geometry or thermal expansion) errors in that orientation. For the second body diagonal a similar measurement is conducted but with the probe head indexed so that the probe is perpendicular to (and offset from) the ram axis as shown in Figure 1. This measurement will produce similar information to that of the first body diagonal position but will include any Z axis roll error in the CMM geometry. In the third body diagonal position, each ball of the ball bar is measured with the probe head indexed in several positions; this will supply information on probe head repeatability and the ability to accurately find a stylus ball center location relative to others in different probe head orientations. In the final ball bar position the ball bar is measured using all the probes in the probe changer rack; this will test for both repeatability of the probe changing process and for any defective probes present. This test, shown in Figure 2, was conducted using a moving bridge CMM and had a floor-to-floor testing time of twelve minutes.



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Figure 2. Results of an interim test using one ball bar in four (body diagonal) positions.

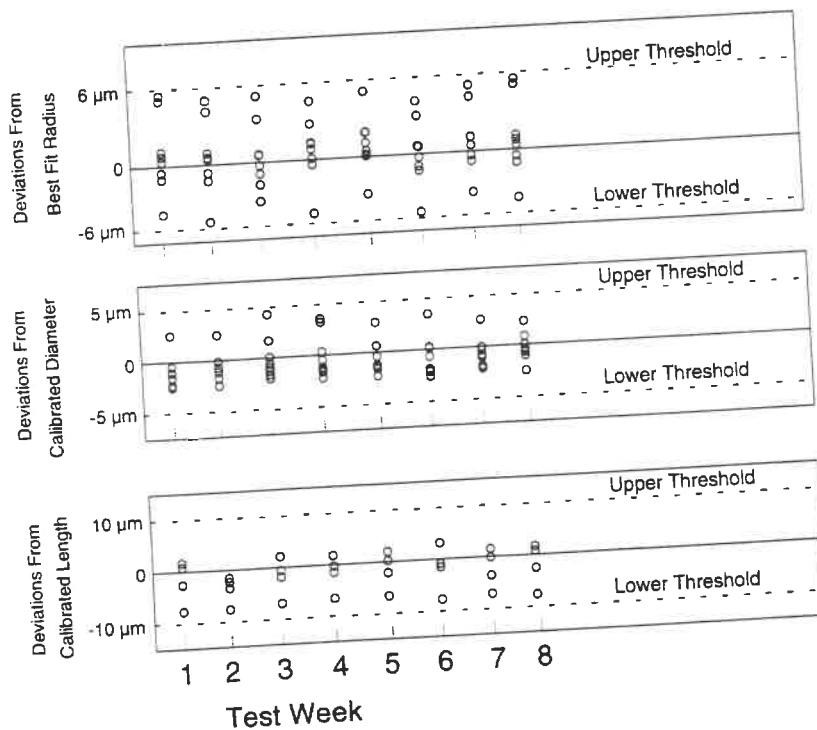


Figure 3: One possible method of summarizing the data from several interim tests. For each interim test all four center-to-center length deviations, all eight ball diameters, and the eight largest radial deviations are plotted. The test is passed if all these measurements are within the threshold value limits which are established by the user.

ERROR CORRECTION MODEL FOR A MACHINING CELL TOMBSTONE

by

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Modern automatic machining cells often use a work piece holding fixture called a "tombstone." The tombstone is a rectangular column made out of steel or cast iron. The tombstone, usually mounted on a machine pallet, allows parts to be loaded and unloaded at stations outside the machining center. The pallet containing the tombstone and work pieces is shuttled into the machining center automatically by the cell controller at the appropriate time. The tombstone concept is an ideal design for automatic handling; however, from the precision engineering view point there are design weaknesses. As precision engineering begins to address automation, new approaches may have to be considered in designing holding fixtures.

Traditionally, when designing very precise machines a great deal of care has gone into making the complete structural loop very stiff. For example, very precise jig bores have used symmetrical bridge structures to support the machining carriage which is stiffer than the "C" type frame machines. The modern machining center, with its work piece holding tombstone, is structurally similar to the "C" frame lying on its back. Designing for automation drives the tombstone toward a smaller column, making it lighter to handle and maximizing the part size which it can hold, but at the same time making it even less stiff.

This paper analyzes the contribution that the deflections of the tombstone makes to the overall error budget of a modern high precision machining center. A model based on this analysis is presented which can be used in an error correction scheme. The error correction scheme is proposed as a method to push the accuracy of modern machining centers toward the ultra-precision regime. The types of errors considered involve both mechanical and thermal deflections of the tombstone. Analysis of the thermal bending of the tombstone has been done. Using coolant temperature data from the machining center at the New Mexico State University Manufacturing Teaching Factory, we predict errors of over 100 microinches from thermal bending alone if the coolant hits one side of the tombstone only. The analysis includes the transient thermal component to predict the changes as the tombstone begins to reach equilibrium with the coolant.

A correction of the thermal bending error can be handled as a z-axis motion adjustment. The correction method involves predicting the magnitude of the correction with an analytical model rather than empirical data. The advantages of the analytical method include an easy change for different types or multiple tombstones that might be used in a machining cell.

Development of a Long Depth of a Focus Optical Microscopy Based on Computer Image Processing

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1. Introduction

Optical microscopes are widely used in many fields of industry as well as for observation of biological specimens. The depth of focus of a conventional microscope becomes shallower as the N.A. of an objective increases, so that observation of 3-D structures in a thick object is generally very difficult.

Optical microscopes are used primarily for assembling the superfine parts of micromachines as well as for performing delicate operations in such as cell fusion, but the limited depth of focus of the microscopes often obstruct these operations, so the development of a microscope system with a greater depth of focus was needed to meet the needs of workers and researchers.

Various techniques have been proposed over the year ¹⁻⁵. They are using a spatial filtering technique, a digital image processing technique and so on.

In this paper the principle of long depth of focus microscopy is proposed. The multiple focal images of the targets of observation are fed into a computer and only the images in-focus selected and synthesized. Four algorithms for making this choice are presented and the corresponding results are shown.

2. System theory

Fig.1 shows basic principle of the picture composition proposed in this paper. Picture images through CCD camera are taken into computer memory at fixed intervals of distance d between objective and specimen. When interval d is set equal to the depth of focus of the optical system, all part of the specimen is taken picture as in-focus picture segments at least once in one of these focal-plane-depth slices (simply as image slices).

Then as shown in Fig.2, each image slice is divided into small processing segments. Focus level is calculated at each processing segment. Focus level of the corresponding processing segments on all image slices, which are the picture from the same part of the specimen, are compared. Because these processing segments observe same part of specimen, computer can compare the focus level without material of specimen or lighting condition take into consideration. By selecting maximum focus level segment at each processing segment, a composite picture image is obtained, and the picture has a depth of focus that corresponds with $d \times$ number of image slices.

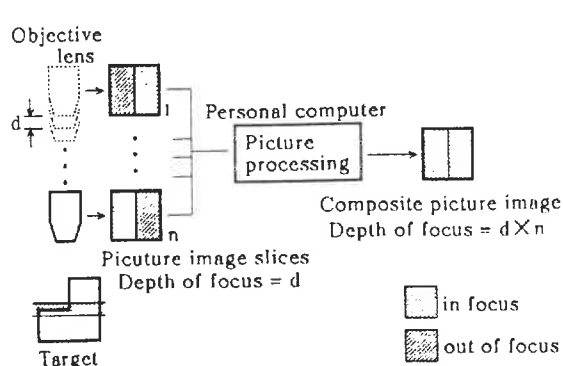


Fig. 1 Theory of expanding depth of focus

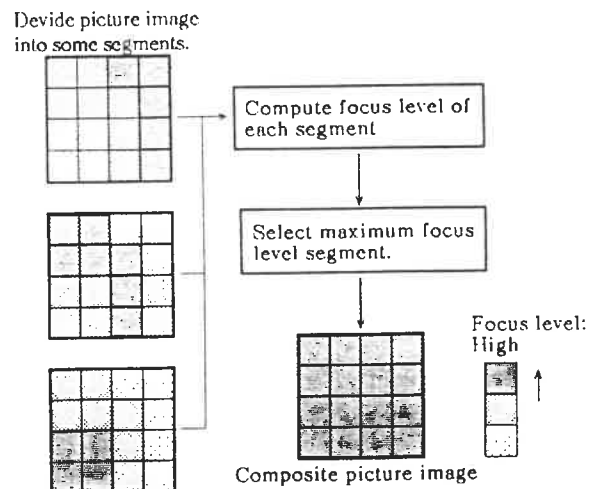


Fig. 2 Procedure of picture composition

A sequence of image slices acquired by CCD camera that is attached to a metallurgical microscope, are transferred to computer through frame memory. One image slice data consists of 512×512 pixels where one pixel has 256 gray levels. A composite image is then assembled by selecting the in-focus segments from these image slices, and shown on the video monitor through frame memory. A piezoelectric microscope focusing unit is used to change gap between objective and specimen. When gaps change larger than $100 \mu\text{m}$ is required, Z-table under the specimen is used, because maximum expansion of microscope focusing unit is $100 \mu\text{m}$.

3. Calculation of focus level

In order to discuss about the focus level calculation algorithm, both in-focus and out-of-focus picture images are prepared. Fig.3 shows various focus level images. A specimen used here is a standard triangular shape surface with amplitude of $1.5 \mu\text{m}$. Z means deviation from in-focus position. The depth of focus is $10 \mu\text{m}$ for the objective used (N.A.=0.21, $\times 10$). The amplitude of surface undulations is small enough to the depth of focus, so that only Fig.3-3 gives in-focus clear picture image. In this study, following four algorithms are presented and corresponding results are discussed.

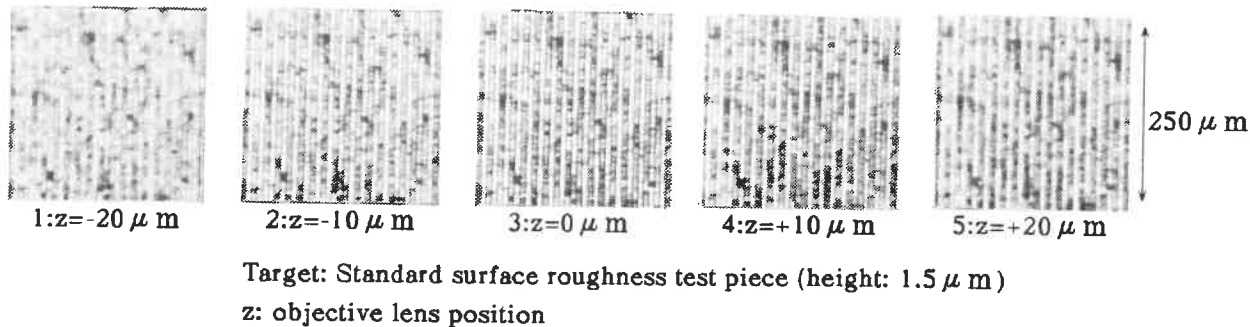


Fig. 3 Samples of various focal-series images

3-1. Difference of brightness method As shown in Fig.3, out-of-focus image slices do not give clear picture images. This is because the difference of brightness of a pixel and its adjoining pixels is smaller for out-of-focus processing segments than for in-focus processing segments. Then a simple method to evaluate the difference of brightness of adjacent pixels is tried. In this algorithm, focus level of a pixel is calculated as the absolute value of brightness difference between the pixel and its adjacent pixels.

3-2. Variance of brightness method Distribution of brightness both from in-focus segment and out of focus segment, are examined. For example, Fig.4 shows histograms of brightness, these are correspond to array of 256×256 pixels of Fig.3-1 and Fig.3-3. From Fig.4 it is obvious that brightness distribution of in-focus segment is wider than that of out-of-focus segment. Therefore variance of brightness can be used as the focus level judgment.

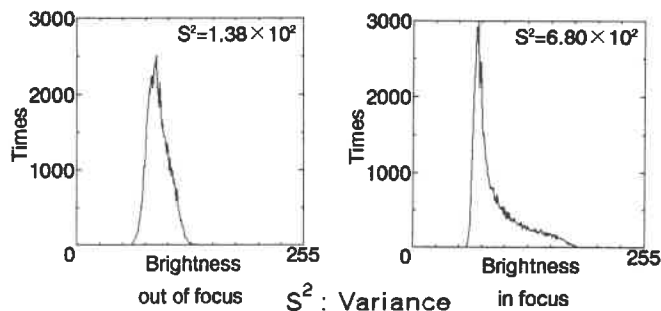


Fig. 4 Histogram of brightness

3-3. Contrast of brightness method As stated above, in out-of-focus processing segments, the difference of brightness at a pixel and its adjoining pixels is small. As a result of this the contrast of brightness, that is, difference of maximum and minimum brightness value in a processing segment becomes smaller as focus level decreases. Fig.4 also shows contrast of brightness at in-focus processing segment is larger than that of out-of-focus one. So, contrast of brightness of processing segment can also be used as a criterion for selecting the in-focus segments.

3-4. Hadamard transfer method Fourth algorithm uses an orthogonal transformation to extract feature of in-focus segments. In this study, Hadamard transformation is introduced as an orthogonal transformation. Hadamard transformation requires short calculation time because its operation consists of only summation and subtraction. By the transformation, array of $N \times N$ pixels is transformed into the sum of $N \times N$ sequency components. Brightness variation of in-focus segment is larger than that of out-of-focus segment. Therefore, the sequency components of in-focus segment must have higher terms.

Quantitative evaluation about the accuracy of these algorithms is the most important point. Table 1 shows the result of comparison of these algorithms. Image composition is carried out for five pictures in Fig.3. The accuracy means the rate of selecting in-focus segments, that is, segments in Fig.3-3 to total segments. Three algorithms except difference of brightness method are applied to segment size 16×16 , 8×8 and 4×4 .

The difference of brightness method requires the longest processing time and the accuracy is not satisfactory. The processing time of the variance of brightness method is short enough, but the accuracy is not so good. The contrast of brightness method has the shortest processing time and the accuracy is also good, but accuracy for small segment size is not satisfactory. The processing time of the Hadamard transfer method is longer than the contrast of brightness method, but it is the best in accuracy.

As a result of this, processing time of view, the contrast of brightness method is the best and accuracy point of view, the Hadamard transfer method is considered the best.

Table. 1 Comparison of the methods of image composition

Methods	Processing segment [pixel]	Processing time [sec]	Accuracy [%]
Difference of brightness	1×1	18.0	58.1
Variance of brightness	16×16	1.08	89.1
	8×8	1.30	80.9
	4×4	2.26	64.6
Contrast of brightness	16×16	0.66	94.9
	8×8	0.82	91.7
	4×4	1.58	79.5
Hadamard transfer	16×16	2.90	99.6
	8×8	3.70	95.4
	4×4	7.40	73.5

4. Results of image composition.

The Hadamard transfer method is applied for actual image composition. Fig.5 shows a CCD camera image of steel plate (0.35 mm thick) on aluminum plate. The N.A. of the objective is 0.21 and the magnification power is 10. The focus is adjusted on the lower plate, the upper plate can not be observed because the step height is 35 times as large as the depth of focus of the optical system. Fig.6 shows the result of image composition by the Hadamard transfer method with processing segments of 16×16 pixels. It is composed from 40 image slices, and individual slice has $10 \mu\text{m}$ interval from each other. Fig.6 shows both upper and lower plate can be observed clearly.

Fig.7 is a CCD camera image for standard surface roughness test-piece (height: $11 \mu\text{m}$). The N.A. of the objective is 0.65 and the magnification power is 40. The amplitude of the profile is 10 times larger than the depth of focus, so the in-focus area is very narrow. Fig.8 is the composite picture image obtained by the Hadamard transfer method with processing segments of 16×16 pixels. It is composed from 20 image slices with $1 \mu\text{m}$ focal series distance. The surface observed in the whole field of view is in-focus.

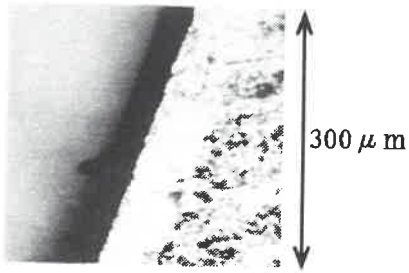


Fig. 5 Steel plate on aluminum plate
(microscope picture image)

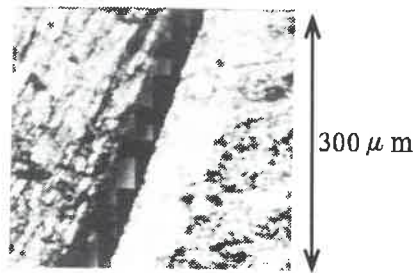


Fig. 6 Steel plate on aluminum plate
(composite picture image)

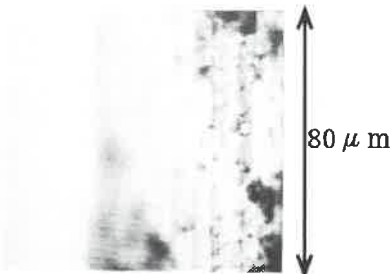


Fig. 7 Standard surface roughness test-piece
(microscope picture image)

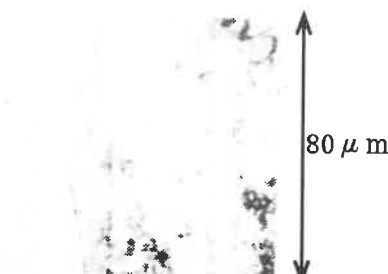


Fig. 8 Standard surface roughness test-piece
(composite picture image)

5. Conclusion

In this paper the principle of long depth of focus optical microscope is proposed. Four algorithms for selecting the in-focus segments were discussed. The results obtained by this research are summarized as follows.

- 1) By the combination of image processing technique proposed in this paper and optical microscope enables observation with arbitrary depth of focus.
- 2) Distinction of in-focus and out-of-focus segments is possible by comparing contrast of brightness or Hadamard transfer image of the picture. The contrast of brightness method has the shortest processing time and the Hadamard transfer method is the best in accuracy.
- 3) The Hadamard transfer method is applied for actual image composition. Specimen, a steel plate on aluminum plate and a standard surface roughness test-piece, were observed in the whole field of view is in-focus.

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Gravitational Distortion of Large Optical Flats

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Key Words: Optical Flat, Surface Contour Metrology, Three Flat Test, Gravitational Distortion, Phase Measuring Interferometer

Most high accuracy phase measuring interferometers measure the flat while it is on edge (vertical). However, if the flat is laid down horizontally, gravitational sag can significantly distort the flat from its "natural" shape. In addition, if the support points of a flat lying horizontally are not kept consistent, the contour of the flat will change, depending on the "natural" shape of the flat and the location of the supports.

Procedures for the measurement of optical flatness are well known, and are described elsewhere.¹ A Zygo Mark IV phase measuring interferometer was used to take all the data presented in this paper, using a standard 10 inch diameter optical flat as the test artifact. This flat was about 2 inches (5cm) thick and was unmounted. i.e., it was a plain, glass blank with both sides polished flat. The test flat had an overall surface figure deviation of about $\lambda/10$, not counting the edge rolloff.

The reference flat used for these experiments was a 13 inch (33cm) diameter, 3.25 inch (8cm) thick glass blank mounted in a kinematic holding cell with a clear aperture of 12 inches (30 cm). This flat is one of a set of three that were purchased with a downward looking system added to the Zygo Mark IV phase measuring interferometer at the AFMSL. They are thicker than is normal for 12 inch diameter reference flats to stiffen them against gravitational sag as much as possible. The test flat was labeled UN1, and the reference flat was labeled DN2. In all cases on the downward looking interferometer, the interference cavity was about 25 to 30 cm deep.

A second "downward looking" flat (DN1) was used to provide a third surface for a three flat absolute measurement of the flats prior to the experiment. In this measurement the reference flat (DN2) was always in the upper cell of the interference cavity, as that is the position that it used during the subsequent tests. This flat, over its central 10 inch area, had a peak to valley deviation on the horizontal and vertical diameter of about $\lambda/50$ (about 12 nm).

The test flat (UN1) was measured

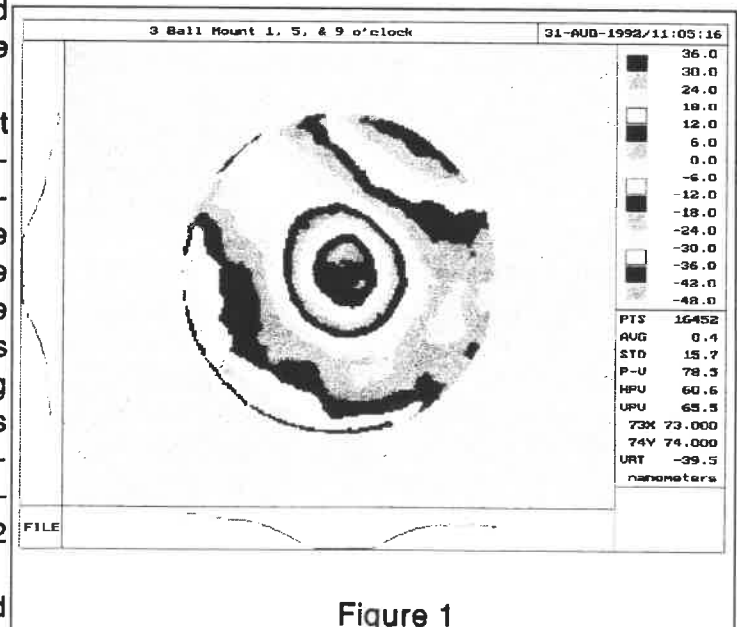


Figure 1

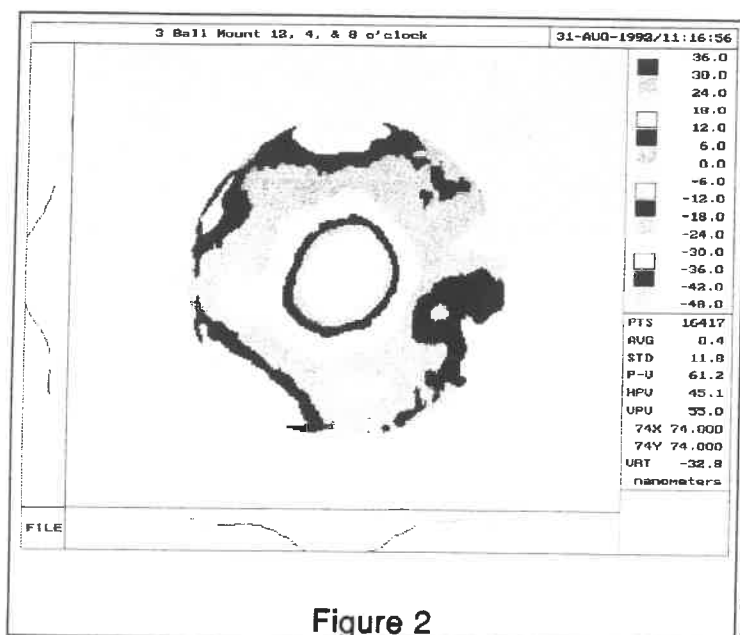


Figure 2

in three different mounting configurations on the downward looking path of the interferometer. First, it was resting directly on a ground steel surface of a rotary table, which is part of the fixturing for the downward looking system. A measurement was made in this position, and high spots can be seen in the contour near the edge of the flat.

Next, flat UN1 was supported on three steel balls evenly spaced on about a 7.5 cm radius and remeasured. Figure 1 is the resulting contour map. Notice that the slope of the flat's surface is relatively steep, running up to the edge at about 1 to 2 o'clock, over one of the mounts, and that there is a less pronounced, but similar slope by the other two mounts. Leaving the mounts stationary, and rotating the flat 90°, (figure 2) we again find high points over the mounting points, even though these correspond to different physical locations on the flat. In the figure, the contour map has been rotated so that it is in the same orientation as in figure 1.

The flat was then set on a soft pad made up of multiple layers of lab tissues with a top layer of lens tissue. This supports the flat evenly over its whole surface, rather than at a few concentrated points. The flat was measured again with the same reference as above and the result is plotted in figure 3. This shape is hypothesized to represent the "natural" shape of the flat. In addition to the central depression, it can clearly be seen that the flat has an overall concave cylinder shape of about $\lambda/20$ with an axis running from about 10 o'clock to 4 o'clock.

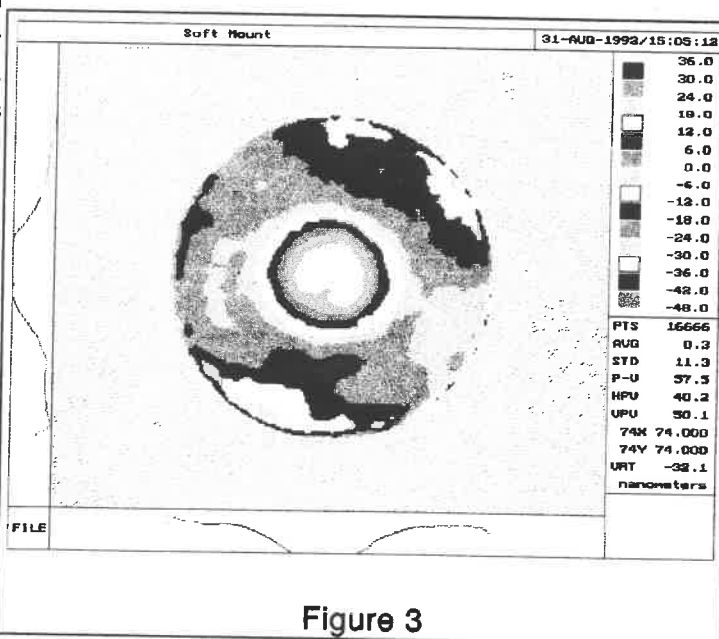


Figure 3

In figures 1 through 3, the change of the flat's surface can clearly be seen under different support conditions. If the contour of figure 3, on the soft pad, is subtracted from that of figure 1, taking care that the differences are performed against the same physical locations of the flat at every point across its surface, we see a change from the "natural" shape of the flat of about 47 nm peak-valley in magnitude. (about $\lambda/13.5$) It can clearly be seen that the high points of this contour correspond to the mounting points of the flat measured and shown in figure 1. Similarly, the difference between figure 2 and figure 3 shows high points that correspond well to the mounting points of the flat shown in figure 2, with a magnitude of 51 nm ($\lambda/12.5$). Taking the difference

between figures 1 and 2, which gives an estimate to the amount of change that can be expected if the mounts are not kept consistent from one measurement to another, we see a peak-valley deviation of this difference plot of about 52 nm, (about $\lambda/12$)

Other attempts have been made to quantify the amount of gravitational sag experienced by large optical flats.^{2,3} (Debenham & Dew, NPL, 1980, and Emerson, W. B, NBS, 1952) In Emerson, the assumption is made that the flat is simply supported by a ring around its outside edge. Debenham & Dew have the flat supported at three points around its edge. While these assumptions allow relatively simple closed form solutions for the deflection under gravity, they do not accurately represent the case described here. Debenham & Dew's assumption can be taken as an approximation, however. Timoshenko & Woinowski-Krieger⁴ give the maximum deflection of a circular plate supported at three points around its edge as:

$$d = 0.0362 \frac{Pa^2}{D} \quad \text{(Eqn. 1)}$$

where D is the flexural rigidity, $Eh^3/12(1-\mu^2)$
P is the total load, $\pi a^2 \rho gh$
a is the radius of the plate
h is the thickness of the plate
ρ is the density of the material
g is the acceleration of gravity
E is Young's Modulus
and μ is poisson's ratio

Substituting and expanding the terms in Eqn. 1 gives:

$$d = 0.0362 \frac{12\pi a^4 \rho g (1-\mu^2)}{Eh^2} \quad \text{(Equ. 2)}$$

Substituting for $a=0.125\text{m}$, $h=0.05\text{m}$, and taking published values for the material properties of glass, we get a maximum deflection due to gravity of about 47nm, which agrees well with the measured values of 47nm and 51nm. This is a somewhat surprising result, considering the difference in the experimental mounting configuration vs the model's assumption. Since the specific type of glass used in the experiment is not known, it is quite possible that its physical properties are sufficiently different to cause this unexpected correlation. It does indicate a reasonable degree of confidence in the result, however.

Notice in Eqn. 2 that the deflection at the center of the flat is proportional to the fourth power of the radius, and inversely proportional to merely the square of the thickness. This means that the thickness must be dramatically increased as the radius increases to maintain the same amount of gravitational deflection. In fact, it quickly becomes impractical to do this for larger flats.

When the flat was mounted vertically, and compared with a standard Zygo transmission flat, we get a figure that looks very similar to the one done horizontally on the soft pad, except that the peak-valley is somewhat larger. (78nm) See figure 4. In this case the reference flat was 12in (30cm) diameter by 2in (5cm) thick flat with a dynaflect coating. This flat is one that is used in production calibration and is itself regularly calibrated via the three flat method. It has an overall concave shape of about 16 nm, ($\lambda/40$) over the vertical "stripe" of the flat, which would tend to increase the peak to valley reading of the test flat with respect to the downward

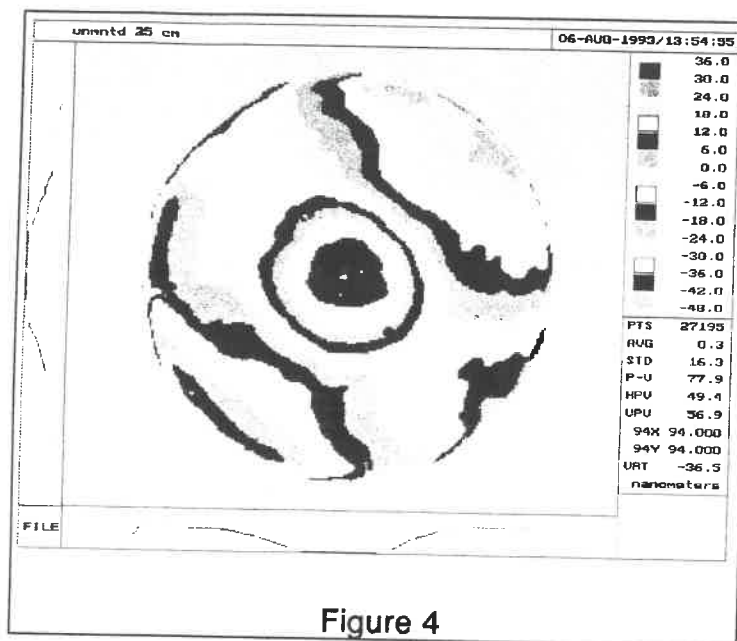


Figure 4

looking case, but not by as much as is seen here, indicating that some amount of distortion may be occurring.

Since the system at the Air Force Measurement Standards Lab does not easily allow the rotation of the 12 in flats on the vertical phase shifter only a vertical stripe of data is measured on the reference flat. Unfortunately, the "downward looking" flat used as the reference for the horizontal measurements cannot be mounted on the edge for simple comparison, due to its size and weight.

Conclusions

While these data showed reasonable comparison between the vertical measurement and the horizontal measurement on the soft pad, some difference is seen between the two orientations. Care must be made in interpreting these results if the mounting configuration of the flat is not kept consistent. For the 25cm flats tested, the best that can be done if the user's mounting configuration cannot be controlled is about 50nm uncertainty. By equation 2 above, the maximum deflection varies as the fourth power of the radius of the flat, and inversely by the square of the thickness. Thus, the magnitude of the gravitational distortion grows quite quickly for larger diameters, requiring the thickness (and weight) to be increased dramatically in order to attempt to compensate. If gravitational sag is not considered, significant errors can be introduced in the as used figure of the flat, as compared to its calibrated value.

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Error Sources in Serrated Tooth Index Tables

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Key Words: Angle Metrology, Index Table, Autocollimator, Optical Polygon

While modern designs for single stage index tables can be quite accurate, care is required during setup and/or calibration to assure that this accuracy is realized. In a series of tests conducted at the Air Force Measurement Standards Lab, a Moore 1440 indexing table was repeatedly measured using a full closure technique under several different conditions of fixturing and environment. These tests indicate that setup errors can be introduced into the measurement that are easily larger than the total error budget of the system.

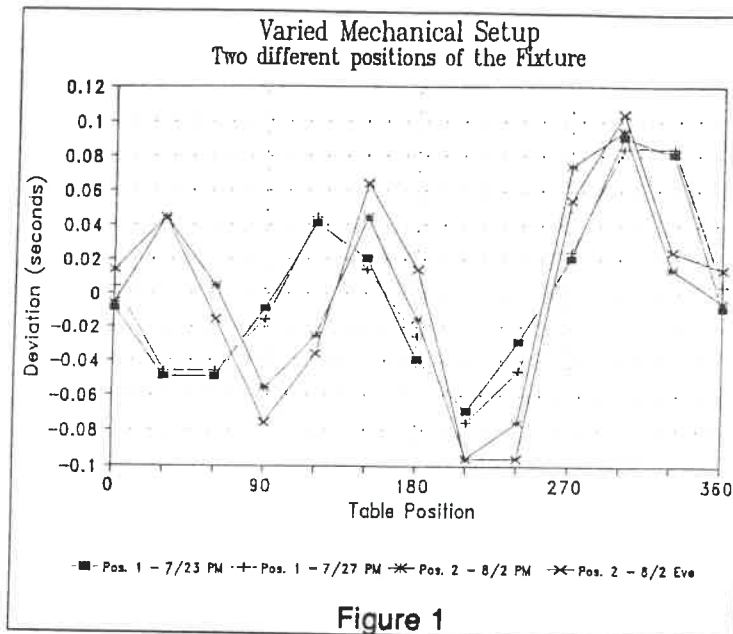
The Moore 1440 Index table is a single stage serrated tooth index table with steps every 1/4 degree. Its overall deviation can be as small as 0.1 seconds of arc "peak to valley" of the deviation plot. The basic design, manufacture, and measurement of single stage serrated tooth tables has been described elsewhere, and will not be repeated here.^{1,2}

Each of the measurements of the index table's deviation was done using a "full closure" design to make the determination of table deviation, using a polygon as the secondary artifact. With this procedure, the necessary data is acquired to allow a solution for all the systematic deviations in both the primary indexing table, and the polygon in an absolute sense. (i.e., without reference to an outside standard.)

Mechanical Setup Variations

In this experiment, the table under test was a Moore 1440 index table, configured with automated drives as part of a polygon calibration station. A second Moore 1440 in this station was used to align the primary table during the course of the measurement. A number of polygons were used in the course of this experiment, including several with 12, 15, 24, and 36 faces. The autocollimator used was a Moller Wedel Elcomat HR, which has a resolution of 0.005 seconds, and an accuracy of approximately 0.02 seconds over small angular changes. The polygons were all solid glass polygons with nominally regular angles, and deviations from nominal on the order of 2 to 3 seconds of arc. Each polygon had a hole through its axis to allow mounting. The polygons were attached to the primary table using a flange shaped fixture that was bolted to the table at three points spaced 120 degrees apart. Two fixtures were used in this experiment, one with a flat base, and one that had relieved pads around the mounting points.

Looking at the previous calibrations of this index table, there is poor consistency of its measured deviations from one calibration to the next. Not only are the overall amounts of the deviations inconsistent, but the pattern of peaks and valleys is not similar either. This, however, is not indicative of the ability of the measurement procedure to determine the table's deviations. Short term repeatability for the measurement is indicated by figure 1. In this figure, deviation plots are shown for two pairs of measurements. Each pair was done without disturbing the mechanical setup of the system in between. Between the two pairs of measurements, however the polygon mounting fixture was removed, rotated 60 degrees, and reattached. While there is excellent repeatability within each pair of measurements, there is obviously a significant difference between the two pairs. Note that the difference between them is up to 0.09 seconds, which is well above the uncertainty of the determination.

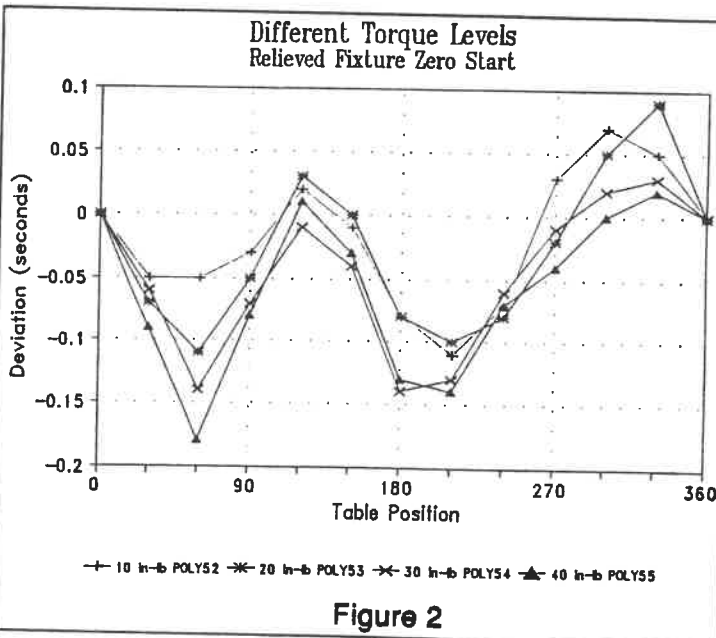


When care is taken to make the setup the same way each time, much more repeatable results can be obtained. Tests indicate that the measurement can be repeated to about 0.04 seconds, if the mechanical setup is done the same way each time. For these tests, the mechanical setup was disassembled and rebuilt in between each measurement. The spread of the data is approximately the width of the uncertainty band of the individual measurements.

It is hypothesized that this change in the deviations of the index table are a result of distortions of the tooth circle of the coupling caused by the mounting stress of the polygon fixture's bolt pattern. One can imagine the tooth

circle of the coupling stretched into an elliptical shape by the mounting stress. In this case the deviation of the table would have a sinusoidal pattern with two complete cycles around the circle, as the upper coupling would match the lower twice per revolution, as the axis of the ellipses goes into and out of "phase" every 180° . This is supported by work done at NIST³, which indicates that the deviations in index table couplings can be effectively modeled by sums of sinusoidal terms with periods equal to even fractions of a circle. ($1/180^\circ$, $1/120^\circ$, $1/90^\circ$, etc.) The "two period" pattern in the table deviations is fairly typical of index tables seen at the AFMSL.

In order to test the above hypothesis, several measurements were performed using different torque settings on the polygon fixture's bolts, with consistent location of the mounting bolt locations on the table's surface. Figure 2 shows the chart of deviations for the table when different torque values are applied to the mounting bolts. The three mounting bolts were tightened with 10, 20, 30, and 40 in-lb of torque respectively. It was found that the torque level of the bolts had a small, but noticeable effect on the value of the table's deviations. In each of these results, the polygon fixture was removed and re-attached between each measurement. The uncertainty of each position deviation is about 0.04 seconds or less, with typical uncertainties at each position being about 0.02 seconds. Especially for the 40 in-lb case, there are apparent differences compared to the other measurements. This torque level is far more than is required to hold the fixture down on the table, and this kind of over tightening is obviously not good metrology practice. 5 or 10 in-lb of



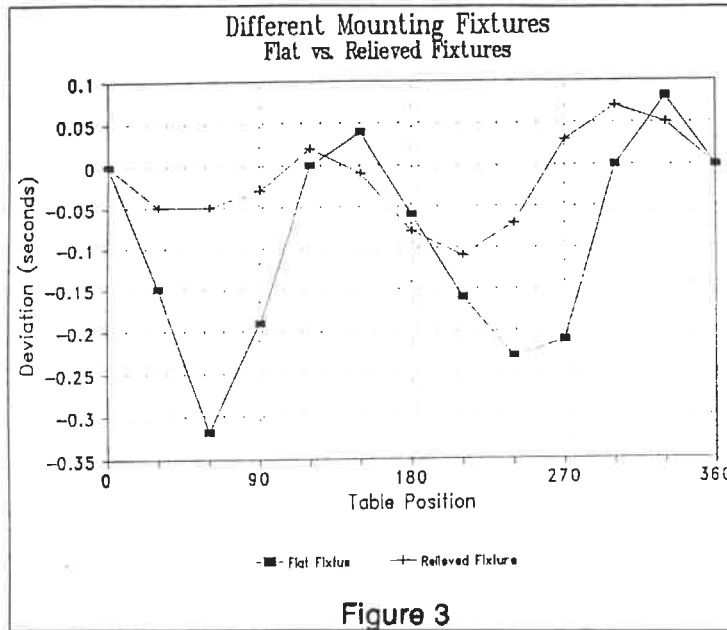


Figure 3

torque on the fastening bolts was sufficient to hold the setup stationary during the measurement.

If the polygon fixture is attached to a different set of bolt holes on the table surface, a significant change of the table deviations occurs. This change appears to be independent of the torque value on the mounting bolts. See figure 1.

The above tests were all performed using a polygon fixture that is relieved on its base. Except for an area about 3/8 inch (10mm) around the bolt holes, the base is cut away so that only the part immediately around the bolt is touching the top of the index table. A fixture with a flat base was also used in this experiment. This fixture is essentially

the same as the first polygon fixture, except that it lacks the relieved base. When this fixture was attached to the same mounting points as the relieved fixture, a significant change occurred in the measured deviations of the table. A comparison of the table's deviations with the two fixtures is shown in figure 3. In this case the difference is almost 0.3 seconds.

Ideally, a kinematic mount of some sort should be used to mount items to the table. For the station used for this experiment, however, it was deemed impractical to design and build such a mount. This station is used primarily to calibrate optical polygons, which are in turn used as angular reference standards for calibrations of items such as rotary tables and encoders. The consistency of the deviation map of the index table used in this station is unimportant for this application, because it will be determined each time the measurement is performed.

Temperature Effects

An opportunity fortuitously arose to test the stability of this index table system under a widely different temperature environment. In May of 1993, work was being done to upgrade the air conditioning system in the Angle Lab. During part of this time the reheat coils of the system were not operating, and the air temperature of the lab was about 60°F (15.5°C) for several days. I took advantage of this opportunity to measure the deviation of the table at this temperature, and compare it to other measurements performed both before and after the environmental system failure.

In figure 4 the results of this

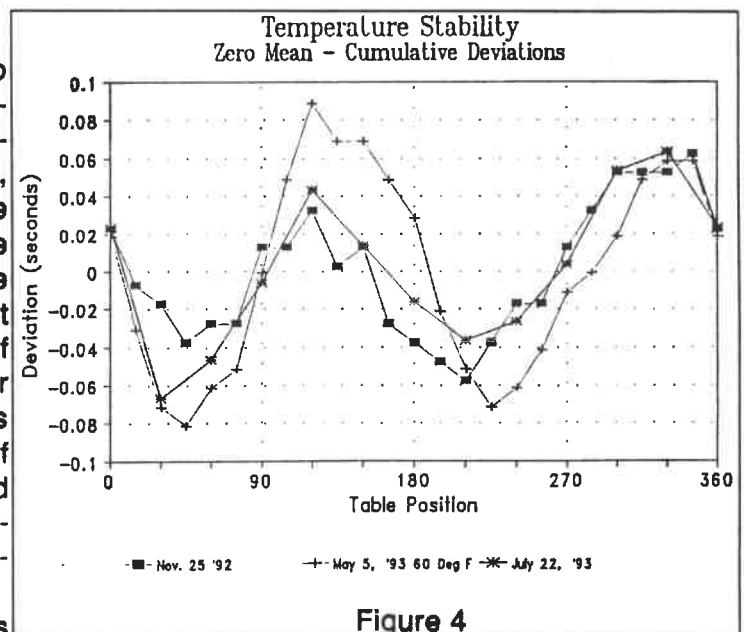


Figure 4

experiment are shown. While there appear to be definite differences in the deviation map for the "cold" measurement, vs. the others, the change is not as dramatic as is seen with the setup variations. However, this test indicates that temperature does cause real effects for high resolution measurements.

Direction of Rotation to Position

In some initial measurements made in the course of this study, it was thought that a difference was seen depending on whether the final position of the coupling was approached from the clockwise or counterclockwise direction. It was thought that this difference could be explained if the settling position of the coupling was different depending on the side of the tooth of the lower coupling that the teeth of the upper coupling slid against to make the final position. Indeed, observing the readout of the autocollimator while the teeth were engaging, it was seen that it would approach its final position from the positive or negative direction depending on whether the table rotation had been clockwise or counterclockwise.

To further test for the existence of a direction of rotation effect, a program was written that alternately moved to the same position from clockwise and counterclockwise directions, and recorded the position in each case. There was no detectable difference in the average position between clockwise and counterclockwise setting. This type of table appears to be quite insensitive to the direction of travel prior to setting.

Further examination revealed that the software used was not taking the data in the proper order in the case where the polygon rotation direction was clockwise. Although all the data were proper for the measurement of the polygon's deviations, the table's deviations were being calculated in a order other than what was reported. When these same data were re-interpreted in the proper order, all differences greater than the measurement uncertainty disappeared.

Conclusions

Consistent setup is the most important factor for achieving accurate, repeatable results using index tables. Even small changes in the set up, such as the difference between a flat based mounting fixture, and a relieved fixture, can cause a change in the measured deviations of the table that are larger than the initial overall magnitude of the deviation. A kinematic mount seems the most desirable method of attaching items to the index table, but lacking this, a permanently mounted face plate to absorb mounting stress would reduce the setup variability. Temperature appears to be a smaller contributor to the lack of repeatability, but can still be important. The direction of travel prior to settling was not found to cause a change in the value of the deviation.

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DESIGN OF A TINY CMM. WITH ULTRA LOW THERMAL EXPANSION GLASS

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1. Introduction

The requirements of the higher accuracy for size and form measurement become nanometer order during several years. The demand for establishing of an ultra accurate Coordinate Measuring Machine (CMM) is urgent in the small size accurate workpieces industry such as micro machine and communication electro - optics.

There are many problems to be solved in increasing the machine accuracy. One of them is a thermal distortion of the construction. Newly developed materials such as super invar, ultra low thermal expansion glass, and carbon fiber are introduced for this purpose 1)2)3). Among them, an ultra low expansion glass has quite an excellent thermal characteristic, however, it is still difficult to apply the actual machine construction due to the weakness of its tensile strength and difficulty of the connecting method for each other. It is believed that if we could overcome such problems, we will be much easy to achieve nanometer level accuracy on precise machines.

In this study, an ultra low thermal expansion glass has been introduced to the main construction of a small size ultra precise CMM, and the design methodology of the glass application have investigated.

2. Basic design

Materials to be used : There are many commercially available ultra low thermal expansion coefficient glass such as Zerodur^(a) , Cervit^(b) , fused silica etc.. Crystron 0 made by HOYA Co., has selected as major component material and fused silica by DENKA Co., has also applied to the base unit. Crystron 0 has similar material components of Zerodur, which mechanical and thermal characteristics are also identical.

Construction style and the capacity : Bridge style construction is one of stable design as well as for thermal deformation and for mechanical vibration. In this case, this design has introduced for increasing the vibration stability. X axis guide has double V design and Y axis incorporates double straight beam design. These elements have symmetrical layout. Total size of the machine is limited by the law material, and decided to within 100 mm travels for X, 50 mm for Y and 25 mm for Z. This size is satisfactory for precision component's industry above mentioned.

Guideway and bearing : The guideway of the X axis is constructed from two 30 mm square straight glass bars with 300 mm length. These bars are placed on the 100 mm thickness fused silica base with V grooves which are ground by diamond wheel. For Y

(a) Schott glaswerke (b) Owens - Illinois

axis, two square bars are placed parallel, and Z axis component is mounted between these bars which is supported by carriage unit. The distance between two bars is precisely kept by the glass blocks which size and surfaces are manufactured to optical quality. Slide mechanism for each axis have dry bearing pads 4), PTFE, these materials are bonded directly on the slide units. Figure 1 shows the friction characteristic of the applied PTFE. . Obtained average friction coefficient is 0.04 for moving condition, and was better than compared composite materials .

Driving mechanism : X table and Y carriage are driven by the DC motor with ball screw thread. Effects of off centering of the ball screw are minimized by the floating mechanism consisted from the auxiliary carriage with 2 spring rods for X axis. Y axis provides the sub-carriage which pushes the center of the main carriage. Inchworm mechanism by PZT is also provided for the Y axis drive for the next step study.

3. Manufacturing

One of the task to be solved for the design with glass material will be the selection of joint mechanism for each component. Several ideas such as bonding, thread rock with insert metal, clamping, and optical wringing could be applied for this requirement. Most accurate method will be an optical wringing but is quite difficult to control the relative position of each other. As a result, the clamping and thread screw jointing method with metal insert have introduced to this development.

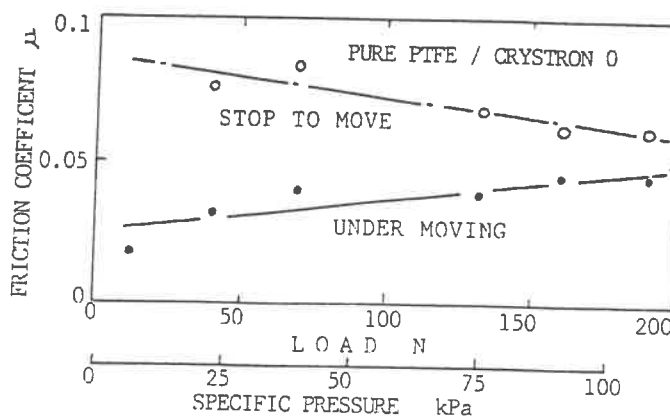


Fig.1. Friction characteristics of PTFE

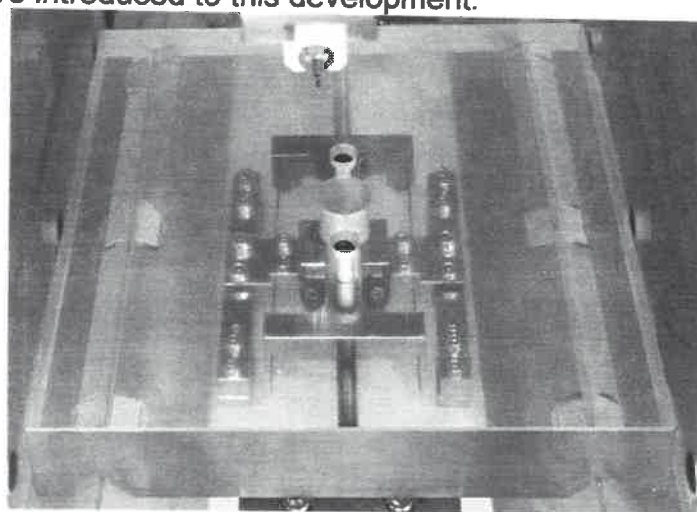
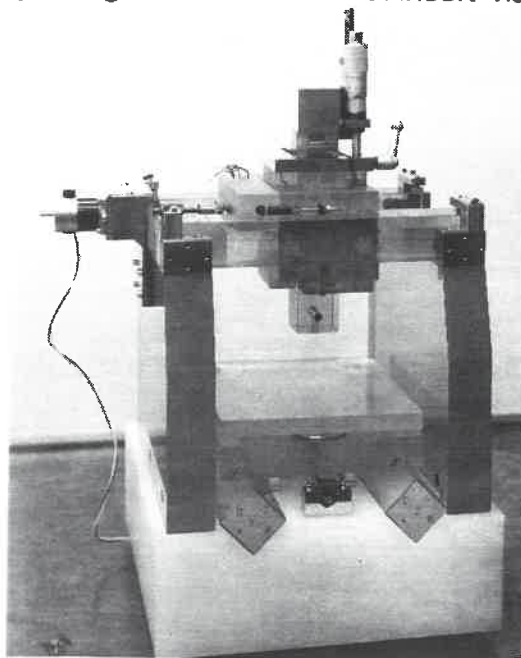


Fig. 3. Close up view of the table driving construction.

Fig. 2 Front view of the developed CMM.

The conventional diamond wheel grinding and diamond powder lapping has been applied to the fabrications of the components. Obtained straightness accuracy of the guide bars after grinding were $1\text{ }\mu\text{m}$ for 300 mm length, and finished to less than $0.2\text{ }\mu\text{m}$ by lapping. Holes were machined by an ultrasonic drill, or diamond drills.

The machine is supported by three polymer vibration isolator SORBOTHANE, and attenuation ratio between floor and stylus could achieve up to 60 dB i.e. $6\text{ }\mu\text{m}$ floor vibrations appear as 6 nm noise level. Appearances of the developed machine have showed in figure 2 and 3. Total appearance is indicated in the figure 2. Figure 3 shows table driving mechanism in the X axis base slide. The ball screw is placed at the center of the base block, and sub carriage which slides on the base block is connected directly to the thread nut. The sub carriage is connected to the main table via two spring wires. This wire isolates the effect of the off center motion of thread screw.

4. Experimental results

Figure 4, 5 and 7 show table travel accuracy. The accuracies were examined by calibrated optical flat and straight edge. From figure 4, X table travel accuracy for vertical direction looks as 20 nm, and optical flat used has 25 nm concave shape which value has calibrated by interferometer, therefore, table travel accuracy seems to be 5 nm in 60 mm travel. Figure 5 shows horizontal directional table travel accuracy of X axis. Obtained result is 50 nm in 100 mm travel. In the figure 7, carriage motion of Y axis has $0.1\text{ }\mu\text{m}$ accuracy in 50 mm travel. Figure 6 shows table travel repeatability for X axis. Small scratch on the optical flat has used for this test, and results obtained are within 1 nm for three times' repeat test.

Transient motion of the X axis table has examined and results are shown in figure 8. The table drive motor operated and stopped suddenly. The figure shows three traces under different kind of table speed. From the result, transient motion seems to be independent from the table speed, and its table floating value is 5 nm approximately which decays with exponential curve.

Squareness between each axis are also measured and obtained values are 1.5 sec. for X/Y and 3.0 sec. for Z/Y.

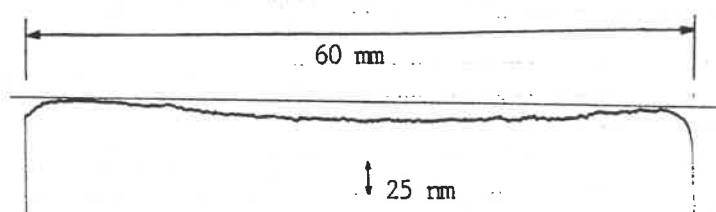


Fig. 4. Obtained X table travel accuracy (Vertical)

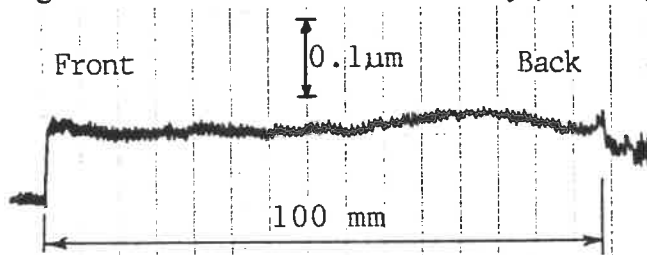


Fig. 5. Obtained X table travel accuracy (Horizontal)

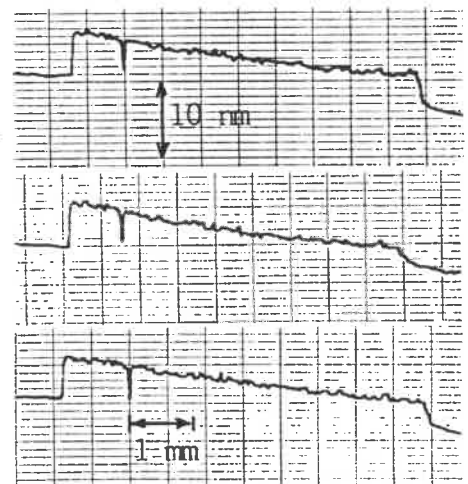


Fig. 6. Repeatability of the X axis table motion(Vertical)

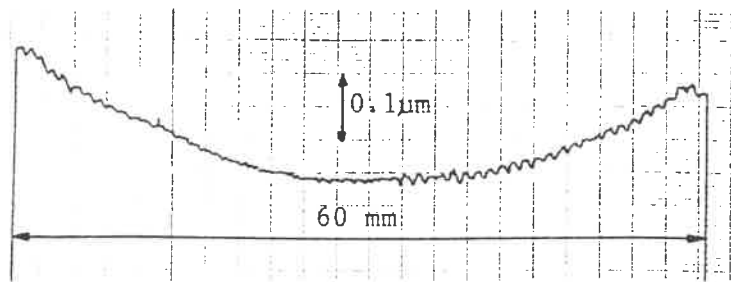


Fig. 7. Y carriage travel accuracy (Vertical direction)

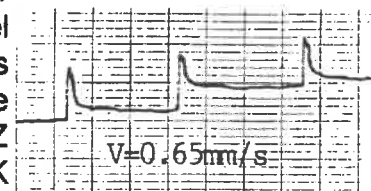
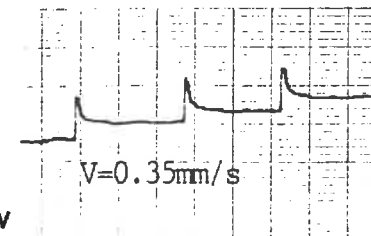
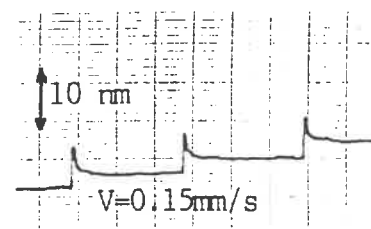


Fig.8 Observed transient motion of X axis table (Vertical)

5. Discussion and conclusion

The developed machine is constructed from ultra low expansion glass and steel parts such as Z axis micrometer screw and ball screw threads of X and Y axis. These steel parts have different expansion coefficient from the glass, this difference generates position error during temperature changing condition. In the Z axis, relative position between Z head and table have changed about $5 \mu\text{m}$ for 10 K temperature change, this value is proportional to the expansion coefficient of the micrometer head.

Y axis accuracy is a little poor comparing to the X axis accuracy, though, its repeatability is within 5 nm. It is possible to improve this error by software compensation. In the Y axis motion, cyclic error motion due to the thread off center motion still remains about 10 nm p-p. The driving mechanism will require further discussion.

Position sensor or probe system development is next task for this program. STM or AFM type probing system will be usable. Also position detector, grating scale or semiconductor laser interferometer and control system are required for next step.

In conclusion, obtained accuracy and anti-vibration results show that such design is effective for a nanometer level ultra precise CMM.

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Laser Rotary Encoders for the Very Large Telescope

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Abstract

Based on the Laser Doppler Displacement Meter (LDDM) Technology and using two laser beams to measure two displacements separated by a fixed distance, the rotational angle which is equal to the difference in displacements divided by the laser beam separation, can be measured. This technique is insensitive to translation, runout or other non-rotational motions. With multiple flat-mirrors or a polygon, the rotational angle of 360 degree can be covered. The resolution is 0.01 arcsec and the accuracy is 0.1 arcsec.

Introduction

In a period of rapid improvement in the capability of telescopes, large telescopes with new mirror designs, adaptive optics and thermal control, may achieve diffraction limited image in the visible region⁽¹⁾. With improved image sharpness and resolution, the demand for pointing and tracking accuracy becomes higher and higher. For example, the Very Large Telescope⁽²⁾ under construction by European Southern Observatory will have 4 separate telescopes with 8.2m diameter mirror each. The resolution in the visible region is 0.17 arcsec. Hence the required pointing and tracking system accuracy is 0.1 arcsec. To achieve such accuracy under heavy load (hundreds of tons) and large temperature variations (tens of degree C) is very difficult.

Conventional techniques, rack-and-pinion, friction drive, Inductosyn, encoder tape, etc. may not be able to achieve the required accuracy. Even using precision steel tape, the errors due to out of roundness and axial run-out are excessive.

Based on the Laser Doppler Displacement Meter (LDDM) Technology⁽³⁾ and using two laser beams to measure two displacements of a single flat-mirror. The rotational angle of the flat-mirror is equal to the difference in displacements divided by the laser beam separation. This technique is non-contact and is insensitive to the translation, run-out or other non-rotational motions of the flat-mirror. Hence the measurement is true rotational angle. Since, the range of the measurement is limited to a few degrees, multiple flat-mirror or a polygon of flat mirrors can be used to cover the rotational angle of 360 degrees.

Described here is the application of the laser rotary encoder for the Very Large Telescope. The major requirements are

Altitude encoder: 1.600m diameter
 Azimuth encoder: 9.820m diameter
 Accuracy: 0.1 arcsec rms
 Resolution: 0.01 arcsec
 Movement uncertainty in all direction, up to 0.5 mm

The basic principle, major features and error analysis will also be discussed.

Laser Doppler Displacement Meter

As described in Ref. 3, the laser doppler displacement meter (LDDM) is based on the principles of radar, the doppler effect and optical heterodyning. Similar to a doppler radar, a target or retroreflector is illuminated by a laser beam. The light reflected by the retroreflector is frequency-shifted by the motion of the retroreflector. The doppler frequency shift can be expressed as

$$f = 2 f_o / C V \quad (1)$$

$$\Delta\phi / 2 \pi = 2 f_o / C \Delta Z \quad (2)$$

Where f and $\Delta\phi$ are the frequency and phase shift, respectively; V and ΔZ are the velocity and displacement of the retroreflector, respectively f_o is the frequency of the laser; and C is the speed of light. A phase detector measures the phase variation, which corresponds to the displacement of the retroreflector.

Using Two LDDM For Angular Measurement

As shown in fig. 1, two single-aperture laser heads are used to measure the rotational angle of a flat mirror. The separation between the laser beam is $d=26$ mm and the distance between the laser heads and the flat-mirror is $L=26$ mm. The rotational angle is equal to $\arctan (\Delta L/d)$, ΔL is the difference between the 2 LDDM readings.

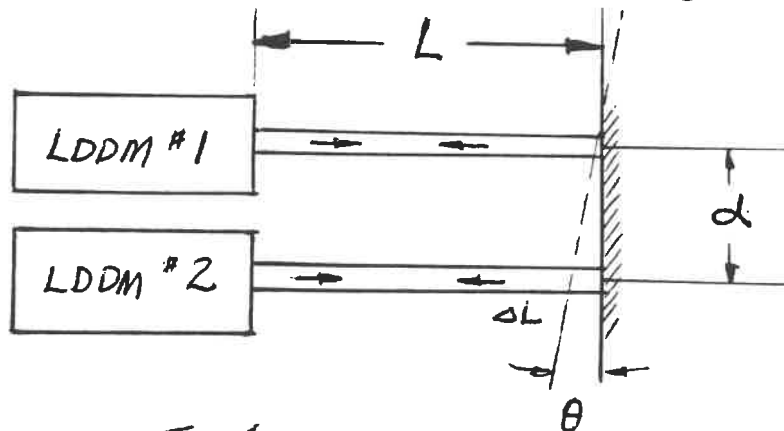


Fig. 1.

For a linear resolution of 1.2 nm, the angular resolution is $1.2 \text{ nm}/26 \text{ mm} = 0.046 \text{ microradian} = 0.01 \text{ arcsec}$.

Since each LDDM laser system (two laser heads) can only measure up to 4 degrees, hence 90 mirrors are needed to cover 360 degrees. Since each system can only measure up to 70% of the rotation angle, two LDDM systems are needed. Let the two LDDM systems separated by 2 degree, then there is at least one LDDM system working during transition from one mirror to another and some of the time both LDDM systems are working.

Major Features

Major features of this method are: measures angle directly; not sensitive to change of center of rotation, non-rotational motion and thermal expansion; large stand-off distance; large alignment tolerance; non-contact; and no back lash.

Error Analysis

The accuracy of the laser rotary encoder system is limited by the rms sum of all errors of each component, namely, flat-mirror, LDDM system, electronics, data transport and computer.

Non-flatness of the first-surface mirror

The flatness of the mirror is specified to be one quarter wavelength.

$$\begin{aligned}\text{That is flatness} &= \lambda/4/25\text{mm} = 6.328 \mu\text{rad} \\ &= 1.26 \text{ arcsec}\end{aligned}$$

Hence, mapping the mirror flatness is necessary to achieve the required accuracy.

LDDM system measurement error

Since the angular measurement is determined by the difference of two linear measurements divided by the separation of these two laser beams, the errors are linear measurements errors and the change of beam separation.

Total measurement Error

Flat mirror:

$$\begin{aligned}\text{Rms total mirror error} \\ \Delta\theta_{\text{mirror}} &= 0.055 \text{ arcsec}\end{aligned}$$

Laser system:

$$\begin{aligned}\text{Rms total laser system error} \\ \Delta\theta_{\text{laser}} &= 0.037 \text{ arcsec}\end{aligned}$$

The total measurement error is the rms sum of mirror error = 0.055 arcsec and laser system error = 0.037 arcsec, Hence
 $\Delta\theta_{\text{total}} = 0.066 \text{ arcsec}$

Assume crossing 56 mirror segments before reset to a reference mark, the mirror crossing error is

$$\begin{aligned}\Delta\theta_{\text{cross}} &= \sqrt{56} \times 0.01 \\ &= 0.074 \text{ arcsec}\end{aligned}$$

Hence the total system error is rms sum of the measurement error and the mirror crossing error, $\Delta\theta_{\text{system}} = 0.1 \text{ arcsec}$

Conclusion

The performance of this rotary encoder has been demonstrated on a 1.6 m diameter ring. It is relatively insensitive to air turbulence and mechanical vibration. The key to the success of this laser rotary encoder system is the use of flat mirrors and the large angular range of the LDDM system.

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DESIGN AND DEVELOPMENT OF AN ENVIRONMENTAL MONITORING SYSTEM FOR THE NANOMETER METROLOGY LABORATORY

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1. Introduction

Nanometer metrology is a branch of nanotechnology, which is defined as the technology where dimensions or tolerances in the range 0.1 nm to 100 nm play a critical role[1]. It occurs to cater for industry's increasing requirements for nanoscale measurement and ultra-precision positioning. Environmental fluctuation can cause dimensional changes and reduce measuring accuracy. It is necessary to control the metrology environment to a pre-determined acceptable level[2]. As a consequence of this demand, a prototype laboratory for nanometer metrology has been built in University of Birmingham, School of Manufacturing and Mechanical Engineering. This was funded through a contract provided by the Research Institute of Science and Technology in Pohang, South Korea.

A controlled environment should possess stable environmental parameters, such as: temperature, humidity, vibration, dust particle count, acoustic noise, etc.. The prototype nanometer metrology laboratory consists of three parts: main laboratory, control room, and basement. It has an air conditioning system which provides constant temperature laminar flow from ceiling to floor over the whole area of the main laboratory. A humidifier adjusts the humidity of the air flow. Furthermore, in combination with two high efficiency particulate air filters (HEPA filters), the air conditioning system is able to efficiently minimise dust particles in the air flow to the laboratory. With respect to vibration control, three large concrete blocks in the basement provide vibration isolation. Considerations of acoustic noise isolation and insulation have been made when the laboratory's walls were designed. It is expected that the laboratory will provide a suitable environment which will ensure that nanometer metrology can be undertaken with reliable measuring accuracy.

2. An environmental monitoring system for the nanometer metrology laboratory

The purposes for setting up an environmental monitoring system for the nanometer metrology laboratory can be presented as follows:

- (1) Initially, the laboratory's capabilities for environmental isolation had to be established in order to assess whether or not the air conditioning system meets the design requirements (with reference to temperature, humidity, and dust particle count). At the same time, vibrations on the three concrete blocks are compared with seismic vibration to identify the vibration isolation capability of the blocks. In addition, acoustic noise level and air pressure in the laboratory should be tested.

- (2) During the working period of the laboratory, the permanent monitoring system plays a role in the management of all of the environmental parameters. If an undesirable situation occurs, the system can issue commands to the related control systems to adjust these parameters.
- (3) Operators can interrogate the controlling computer which is monitoring the environmental situation without the necessity of their entering the environmentally controlled laboratory. Thus, operators can make decisions about whether or not the current environmental situation is suitable for their specific precision experiments according to the information obtained from the monitoring system.

3. Design of a computer-based environmental monitoring system

A monitoring system normally includes three parts, (a) sensors to collect the physical characteristics, (b) signal processing to convert the physical signals into electrical signals which can be processed into readable forms, and (c) information output to provide monitored results. The physical characteristics monitored by the environmental monitoring system are environmental parameters, such as: temperature, humidity, vibration, air pressure, dust particle count, etc.. A microcomputer was selected to manage the monitoring system. It performs: data acquisition from an A/D converter (RTI-815), data processing, and information output to both the computer's screen and ASCII data files on the computer's hard disk.

A "one-to-one" monitoring model was adopted for the system. This requires the monitoring of a parameter that is a measure of some particular characteristic with a suitable sensor. This is a "one-to-one" configuration in which one sensor relates to one physical characteristic. This monitoring model is convenient in use and flexible in operation.

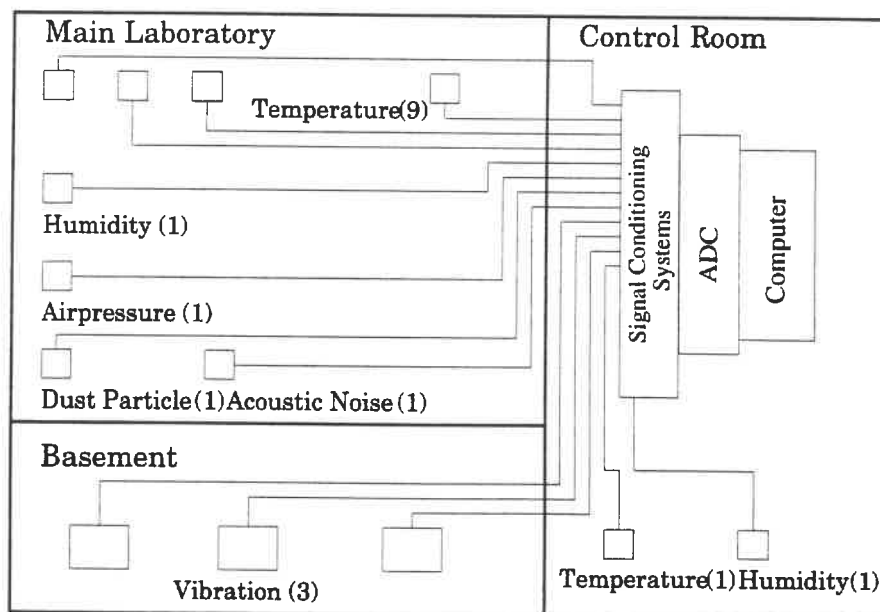


Fig 1. Schematic distribution of the monitoring system

Figure 1 presents a schematic diagram of the monitoring system's distribution. In order to avoid generating noise and inputting heat to the laboratory, all of the signal processing systems and the microcomputer itself were located in the control room. Only the sensors were installed in the main laboratory. Thus, temperature, humidity, acoustic noise, air pressure, and dust particle count were interrogated by the monitoring system. Each sensor was mounted in a reasonable position, and linked with the microcomputer's interface in the control room through a screened cable. Since the laboratory was designed to have downward laminar flow, there were several temperature sensors mounted in the main laboratory to test the temperature at different levels and also to detect the change in temperature at the same level. In addition, one humidity sensor, one air pressure sensor, one sound level meter, and a dust particle count device were placed in the main laboratory to carry out related monitoring tasks. The control room also has a temperature sensor and a humidity sensor. With respect to vibration isolation, accelerometers were installed on the three concrete blocks in the basement to identify their capability of vibration isolation. Vibration measurement could be undertaken in both vertical and horizontal directions.

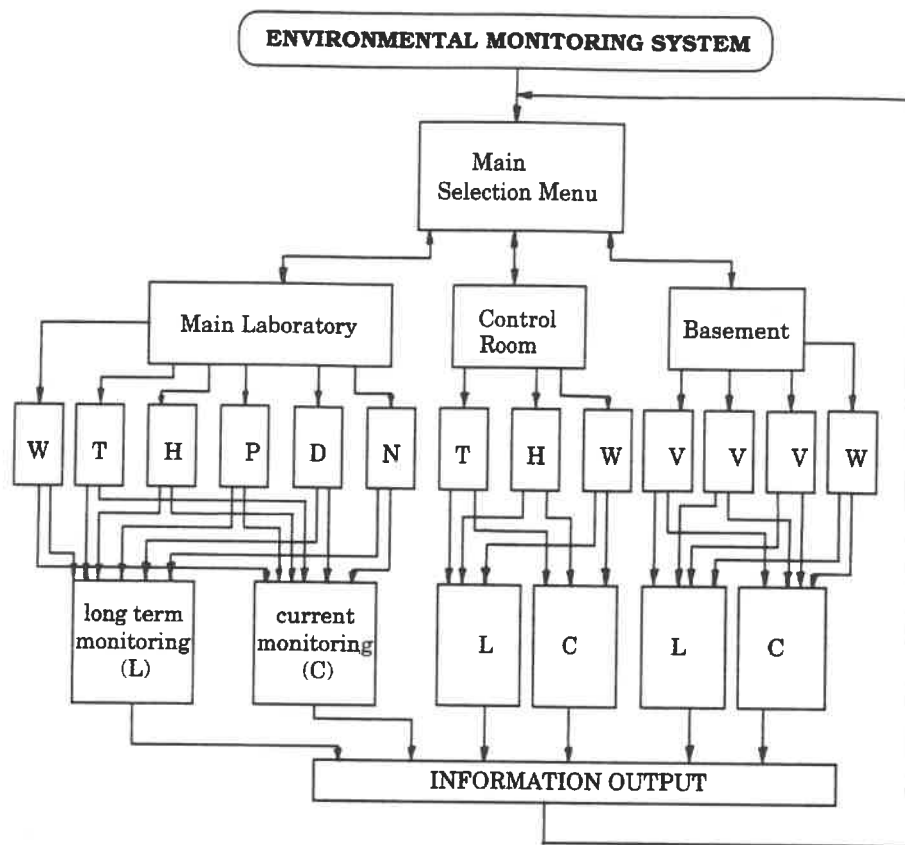
4. Development of controlling software for the environmental monitoring system

The software used in the environmental monitoring system performs: data acquisition, data processing, and data file management. C++ was selected to achieve the purposes. The design criterion for the monitoring software was that it should be user-friendly. Since communication between operators and the microcomputer is necessary, the software should have a suitable mechanism for communication. During data acquisition and data processing, the measurement accuracy should in no way be reduced by the operation of the software. Especially for high frequency signals such as those associated with acoustic noise and vibration, the speed of data acquisition should match the speed of signal change.

Figure 2 shows the software design flow chart. In the software system there are several menu windows to provide the operators with various alternatives. In the nanometer metrology laboratory, there are several environmental parameters needing to be monitored at each room (the main laboratory, the control room, and the basement). The main selection menu is provided for the selection of a specific room where environmental parameters are required to be monitored. The second selection menu allows the selection of a specific parameter to be monitored, such as temperature, humidity, etc.. At the same time, there is an item for current overview of all of the parameters in the same room. The sub-menu under each parameter includes two selections for showing the long term monitoring or the current parameter situation. Finally, various information output windows are available through the microcomputer's screen to demonstrate the monitored results by means of graphs and text. Simultaneously, for long term monitoring, a data file can be generated and stored on the computer's hard disk as a record. The software is able to return to the main menu from every selection menu or when a monitoring chain is completed, whereupon it then awaits the operator's commands. Operators can decide to quit the programme and return to DOS environment at any time.

The software that comes with the RTI-815 board provides various data acquisition routines, which are supported by a Borland C++ compiler. The Nyquist interval indicates that the

sampling frequency should be at least twice the signal frequency[3]. Considering a 25 μ s conversion speed of the A/D converter, its maximum sampling frequency can theoretically match a data acquisition speed of 20 KHz for random signals. This is an acceptable speed for the acquisition of vibration and acoustic noise signals.



W-whole parameters; T-temperature; H-humidity; P-air pressure; D-dust particle;
N-acoustic noise; V-vibration

Fig. 2 Overview of the software flow

5. Conclusions

The computer-based environmental monitoring system has been in service in the nanometer metrology laboratory at Birmingham University for approximate one year. Its performance has demonstrated that the one-to-one monitoring system is successful for environmental control, and that the software system is satisfactory in use.

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Uncertainty Analysis of Circle and Arc Measurement Using Discrete Sample Points

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INTRODUCTION

In recent years, we have seen the convergence of using computer controlled coordinate measuring machines (CMMs) for dimensional measurement of precision manufactured parts. Coordinate metrology has emerged as an important area in precision engineering and demands special attention to address some of the vital issues regarding quality control and assurance. One of these issues evolves around the method divergence of measurement evaluation [1]. In coordinate metrology, discrete sample points are used to construct substitute geometry and evaluate geometric tolerances. Method divergence, therefore, refers to the fact that different results are produced by different methods for the same geometry. Among factors that affect the evaluation results, sampling strategy and best fit algorithm are two of the most critical ones. Although research [2] has been conducted to investigate sampling strategies for circles in coordinate measuring machines, only limited empirical knowledge was obtained based on computer simulations. There is no theoretical model to accurately predict the uncertainties of the evaluation results.

The work reported in this paper addresses the uncertainty analysis of geometric evaluation using discrete sample points. When applied to the evaluation of circles and arcs, analytically it is clearly shown that the uncertainties associated with the assessed center coordinates and diameter is a function of the number of sample points, their geometric locations, and the dimensional errors produced by the manufacturing process.

UNCERTAINTY ANALYSIS

Given a set of discrete measurement points for a geometric feature, the parameters of a substitute geometry are usually evaluated by certain best fit algorithms (e.g., least squares method). If we define deviation in the surface normal direction as the geometric error (which includes measurement error and dimensional error), we can relate the parameter errors to the geometric errors in a compact matrix form:

$$\tilde{\epsilon} = A \delta \tilde{t} \quad (1)$$

where A is the Sensitivity Matrix [3]. For example, equation (1) can be expanded for a 2 D circle as:

$$\begin{bmatrix} \epsilon_1 \\ \epsilon_2 \\ \vdots \\ \epsilon_m \end{bmatrix} = \begin{bmatrix} n_x^1 & n_y^1 \\ n_x^2 & n_y^2 \\ \vdots & \vdots \\ n_x^m & n_y^m \end{bmatrix} \begin{bmatrix} \delta x \\ \delta y \end{bmatrix} \quad (2)$$

where ϵ_i is the geometric error for the i th point, $[n_x^i, n_y^i]$ the unit normal, m the number of points, and $[\delta x, \delta y]$ the offset error of the circle center. Assuming t_i and ϵ_i are normally distributed random errors with zero means and let the sensitivity measure of t_i to ϵ_i be defined as

$$S_{t_i} = \frac{\sigma_{\epsilon_i}}{s} \quad (3)$$

where $\sigma_{t_i}^2$ is the variance of δt_i and s^2 is the variance of ϵ_i , a general sensitivity equation can be derived [3] as:

$$S_{t_i}^2 = \sum_{j=1}^m [(A^T A)^{-1} A^T]_{ij}^2 \quad (4)$$

This sensitivity measure relates the uncertainty of the best fit transformation parameters to the geometric errors and the local attributes of the measurement locations.

Equations (4) is general to all geometric features. It can be evaluated given the measurement locations and their normal vectors. To show the significance of this result we use a 2 D circle with uniform angular spacing of sample points as an example. According to equation (2) and (4),

$$(A^T A)^{-1} A^T = \frac{2}{m} \begin{bmatrix} n_x^1 & n_x^2 & \dots & n_x^m \\ n_y^1 & n_y^2 & \dots & n_y^m \end{bmatrix} \quad (5)$$

Therefore,

$$S_{t_x}^2 = \left(\frac{2}{m}\right)^2 \sum_{i=1}^m (n_x^i)^2 = \frac{2}{m} \quad (6)$$

$$S_{t_y}^2 = \left(\frac{2}{m}\right)^2 \sum_{i=1}^m (n_y^i)^2 = \frac{2}{m} \quad (7)$$

From equations (3), (6) and (7),

$$\sigma_{t_x} = \sigma_{t_y} = \sqrt{\frac{2}{m}} s \quad (8)$$

Equation (8) states that the uncertainty of t_x and t_y is proportional to the magnitude of geometric error and is inversely proportional to the squared root of the number of points. In other words, as the variance of geometric errors increases, the uncertainty of the evaluated center position of the circle increases. On the other hand, as the number of sample points increases, the uncertainty decreases. It is interesting to note that it is independent of the size of the radius. However, the uncertainty of the diameter evaluation can be similarly related to the geometric errors [3] as

$$\sigma_D = \frac{2}{\sqrt{m}} s \quad (9)$$

To verify the results, simulations have been carried out with different number of uniformly distributed measurement points superimposed by a random error produced by a Gaussian random generator. The magnitude of the standard deviation of the random error is one micron. The number of points spans from 3 to 20. For each number of sample points, one thousand simulations were collected to calculate the standard deviation of the evaluated parameter by the least squares method. The predictions based on equations (8) and (9) were also calculated. Table 1 shows the simulation results. Clearly the predictions and the actual numbers match quite well. Figure 1 graphically shows the trend as the number of sample points increases.

Equations (8) and (9) can also be used to determine a minimum number of measurement points when the circle is toleranced with size or true position. This is especially important for CMM inspection, since the uncertainty introduced to the evaluated diameter and center position using just few number of

sample points is quite significant. Given the known manufacturing process and process variability (which can be used to predict S), one can estimate the minimum number of points that is required to control the uncertainty to be consistently less than a fraction of the tolerance (usually 10% to 30%).

If we generate simulation data on an incomplete circle, we are simulating the measurement of an arc. For the uncertainty analysis, we no longer use equation (8) but equations (3) and (4) directly. The number of points is fixed at ten and the arc angle is progressed from 10 to 360 degrees (symmetric to the Y axis). Figure 2 shows the predicted standard deviations of δx and δy . Note that this uncertainty analysis does not consider the error induced by algorithms. In other words, in the case of using a perfect algorithm, the uncertainty still exists and follows the pattern in the figure.

SUMMARY

In short summary, uncertainty of the evaluated best fit parameters can be related to dimensional errors, number of measurements, and their corresponding locations. The uncertainty analysis is general to all geometric features since it does not require geometric parameters but only position and normal vectors to be known. Equation (4) can be easily programmed to calculate the sensitivity measure of the transformation parameters to geometric errors. Using this measure, one can evaluate the uncertainty of the best fit transformation and control the consistency of the analysis results.

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Table 1 Uncertainty estimation (calculated sigs) compared with average "actual" values from simulations.

NUMPT	sig_x	cal sig_x	sig_y	cal sig_y	sig_D	cal sig_D
3	0.000812	0.000816	0.000811	0.000816	0.001130	0.001155
4	0.000739	0.000707	0.000714	0.000707	0.001008	0.001000
5	0.000638	0.000632	0.000642	0.000632	0.000859	0.000894
6	0.000571	0.000577	0.000593	0.000577	0.000806	0.000816
7	0.000527	0.000535	0.000525	0.000535	0.000752	0.000756
8	0.000500	0.000500	0.000495	0.000500	0.000723	0.000707
9	0.000461	0.000471	0.000455	0.000471	0.000673	0.000667
10	0.000441	0.000447	0.000456	0.000447	0.000618	0.000632
11	0.000425	0.000426	0.000432	0.000426	0.000613	0.000603
12	0.000417	0.000408	0.000398	0.000408	0.000560	0.000577
13	0.000418	0.000392	0.000394	0.000392	0.000539	0.000555
14	0.000397	0.000378	0.000370	0.000378	0.000540	0.000535
15	0.000380	0.000365	0.000370	0.000365	0.000514	0.000516
16	0.000366	0.000354	0.000354	0.000354	0.000504	0.000500
17	0.000327	0.000343	0.000341	0.000343	0.000479	0.000485
18	0.000337	0.000333	0.000335	0.000333	0.000476	0.000471
19	0.000335	0.000324	0.000327	0.000324	0.000463	0.000459
20	0.000303	0.000316	0.000315	0.000316	0.000432	0.000447

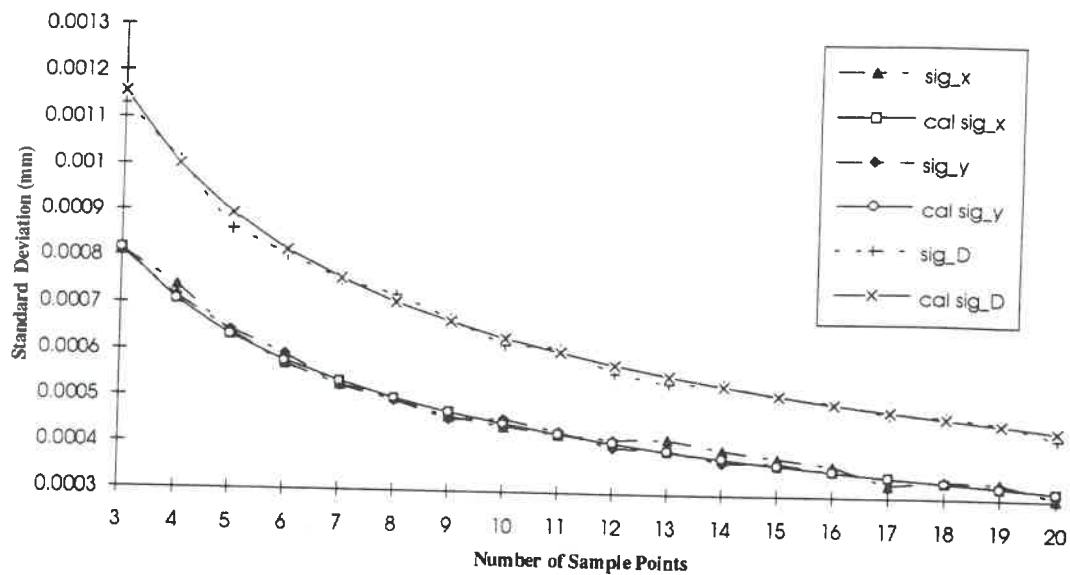


Figure 1. Uncertainty predictions and actual values vs. number of sample points.

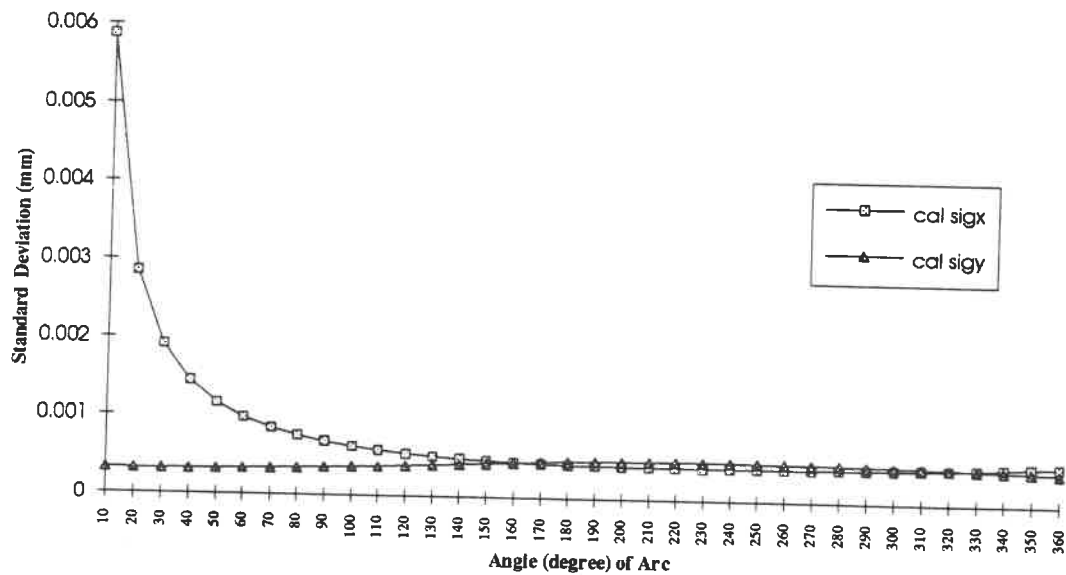


Figure 2. Uncertainty predictions versus arc angles.

SURFACE TOPOGRAPHY ANALYSIS AND REPRESENTATION FOR AUTOMATED SURFACE FINISHING PROCESSES

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Production of molds and dies is usually labor-intensive, thus very expensive, and may involve very long lead times. For example, an injection mold cost of \$50,000 is rather typical and some molds may cost in excess of \$200,000. Surface finishing accounts for up to 50% of the total time required to produce a mold or die, and most finishing operations are still performed manually [1]. Automation of surface finishing is imperative to reducing labor content and overall cost while producing higher quality manufactured parts, and integration of in-process surface topography measurement is a key to automated surface finishing. In the past there has been no suitable surface topography representation and surface inspection data analysis method [2-6] that could be used in feedback control of surface finishing processes, simultaneously addressing the more macro aspects of shape error and finishing-machine motion generation, and the more micro aspects of waviness and roughness removal.

Figure 1 shows the block diagram of surface finishing process integrated with surface topography analysis and representation that has been developed at the University of Wisconsin-Madison. The term "surface topography" is defined here to encompass the surface shape, surface waviness, and surface roughness [7] of a workpiece and the term "surface finishing" is used to encompass modifications made in workpiece topography that are relatively small compared to the dimensions of the surface of the workpiece. The desired surface topography is a given specification from the CAD database. A method of surface topography analysis is described in this paper to efficiently separate the surface shape error and surface waviness after filtering out the surface shape and finding surface waviness elements. Surface roughness has been analyzed effectively by the same method. An approach of surface topography representation has also been developed to represent the surface waviness by the B-spline fitting method and can be directly combined with the process mechanics model for controlling surface finishing processes.

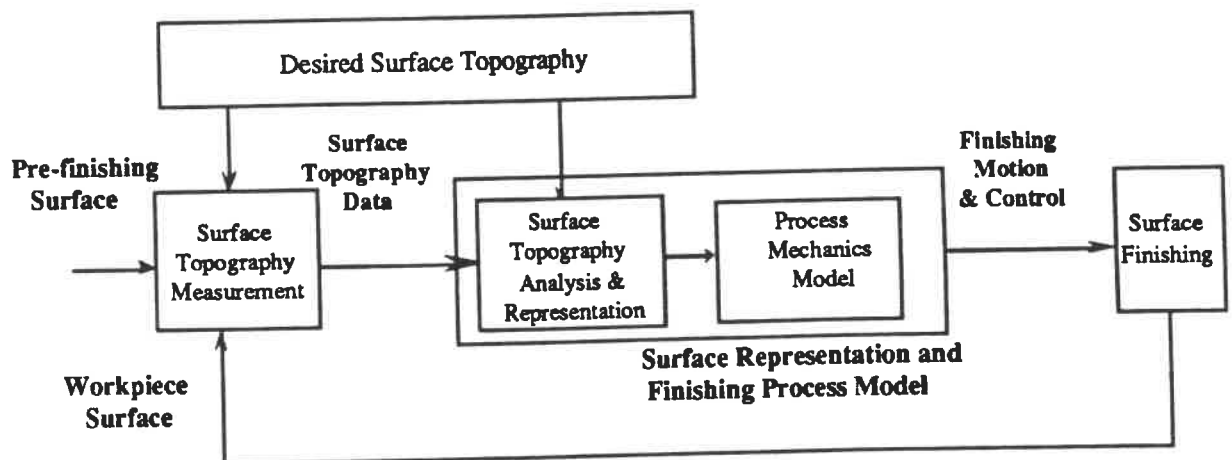


Figure 1. Finishing process with surface topography analysis and representation

SURFACE TOPOGRAPHY ANALYSIS

Surface topography analysis has been developed in two steps: (1) surface shape filtering; and (2) motif filtering for surface waviness and roughness. The purpose of surface topography analysis is to separate the shape error, surface waviness, and surface roughness from the inspection data, assuming that the acquired surface data contains enough surface topography information. Shape error is the deviation of the machined shape from the desired surface shape as shown in Figure 2 [8]. The mean line of the machined profile is commonly accepted as the machined shape [7]. However, it is difficult to find the mean line of whole machined profile due to the surface data representing the superimposition of surface shape, surface shape error, surface waviness, and surface roughness. Thus, the first step of surface topography analysis is to use the desired surface function as a surface shape filter, the output of which represents residual waviness and roughness errors with respect to a flat surface in a 3-D domain, or a straight line of surface profile in a 2-D domain. For 2-D surface profile analysis, Figure 3 shows a waviness element that is defined as a block containing two peaks and one valley. Each element is characterized by its length and two heights. Instead of finding the mean line of whole machined profile, the arithmetic average of heights of inspection points inside one waviness element is calculated when the construction of waviness elements is completed. Shape error, e_i , is then obtained for each waviness element as shown in Figure 3.

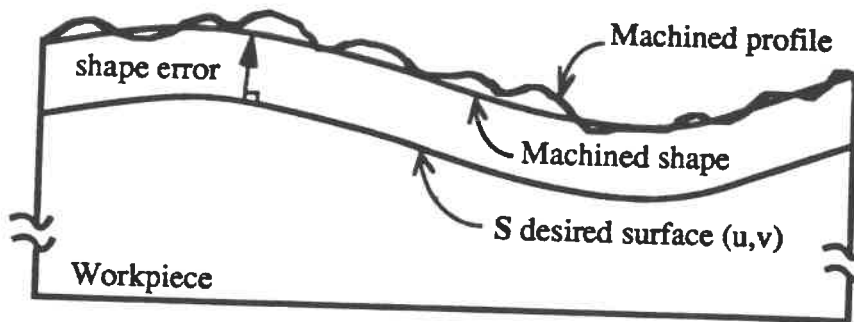


Figure 2. Surface shape error and desired surface

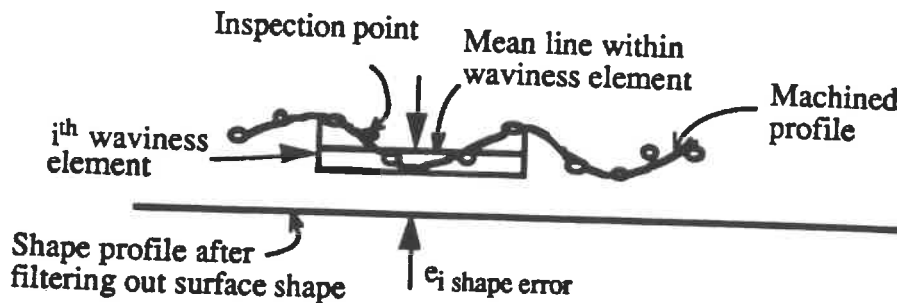


Figure 3. Definition of waviness element and surface shape error after shape filtering

After surface shape filtering, a motif filter [9] with optimal cut-off length is used to analyze the surface waviness, and surface roughness from the surface topography information. Based on the repeated geometric properties of peaks and valleys in inspection data, four combination rules (envelop, width, enlargement, and relative depth) are used for finding waviness motif elements and roughness motif elements from the inspection data. A threshold value of 2.5 mm was used in the width rule for waviness and 0.5 mm for roughness [9]. Figure 4 (a) is the plot of surface

topography data obtained from a flat, fine-milled surface sample inspected using an optical probe installed on a CMM. The inspection data for one profile pass in the X-direction consisted of 365 points and the inspection interval was 0.05 mm. For illustration purposes, Figures 4 (b) and (c) show results of surface data analysis of one section which contains 165 points in one profile pass. In Figure 4 (b), all of the peaks and valleys of the original data were found first. Waviness motif elements were then constructed by combining or dividing according to the four combination rules of testing the waviness height (W_i) and wavelength (AW_i) of each waviness motif element. Roughness motif elements were obtained by the same rules from the roughness height (R_i) and wavelength (AR_i). The result of construction of roughness motif elements is shown in Figure 4(c). This motif filter method is efficient in processing a large amount of data because there is no iteration and transformation in the four combination rules. This surface topography analysis appears to be a good candidate for analyzing raw surface topography data after surface shape filtering, and separating the waviness and roughness components in a way that is suitable for input to process models on a macro and micro scale and subsequently producing required finishing motions.

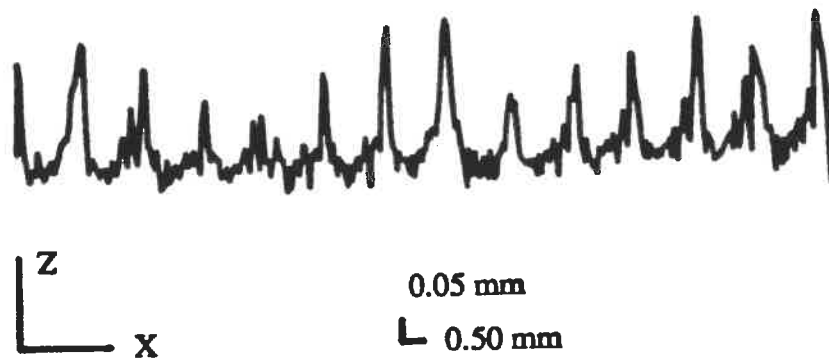


Figure 4 (a). Surface topography data

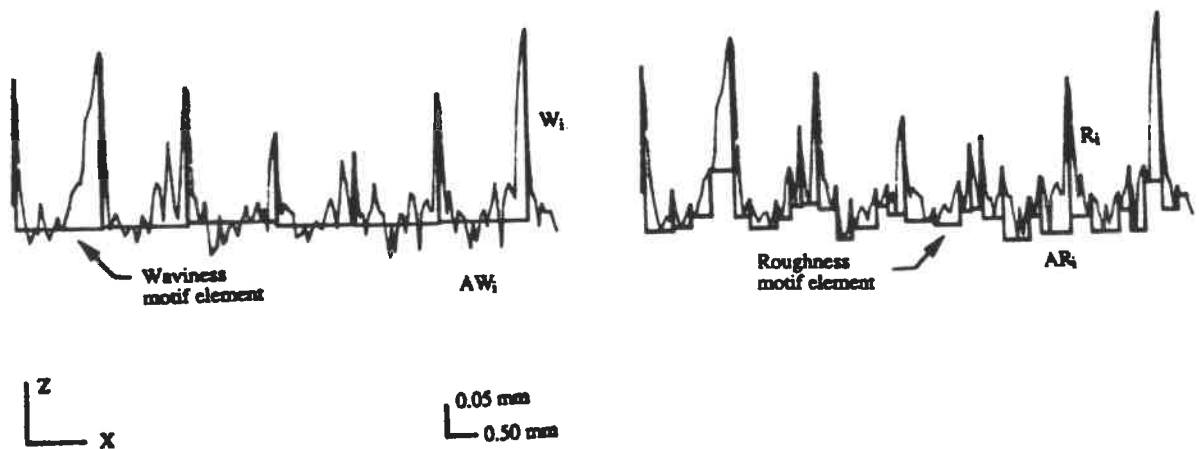


Figure 4 (b). Output of motif waviness filter

Figure 4 (c). Output of motif roughness filter

SURFACE TOPOGRAPHY REPRESENTATION FOR SURFACE WAVINESS

After constructing the waviness motif elements, a B-spline fitting model [10] is used to represent the surface topography for obtaining the control tracking of finishing motions. With multiple knots and four fitting polygon vertices, a cubic B-spline curve is chosen here to fit the waviness data in each waviness motif element. The result of test sample waviness represented by B-splines is shown in Figure 5. Multiple knots are used to control the continuity and the four fitting polygon vertices are adjusted for each waviness element. Advantages of using B-splines to represent waviness are: (1) the control of finishing tools is based on a continuous waviness function instead of the discrete inspection points; and (2) the B-spline mathematical representation is compatible with and can be easily merged into CAD surface shape profile representations. Therefore a conventional CAD/CAM system can be used to generate surface shape error correction and waviness removal finishing motion commands for surface finish processing equipment.

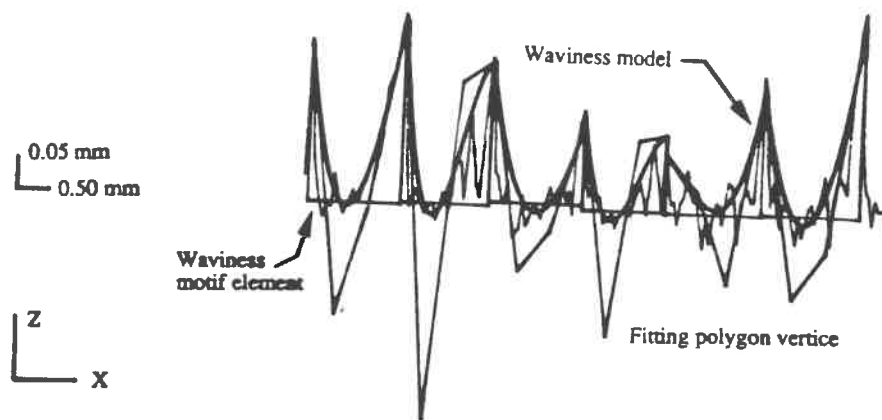


Figure 5. Surface waviness representation by B-spline fitting

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High-speed non-contact profiler based on scanning white light interferometry

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Scanning white light interferometry (SWLI) [1-6] is a technique that uses interference from broad-band illumination to measure relative distances and can be used to profile surfaces with much larger discontinuous departure than conventional phase shifting techniques. By axially scanning one leg of an interferometer and analyzing the resulting 3 dimensional interferogram, a profile of the surface illuminated at the other leg can be reconstructed. We describe a new profiler based on SWLI techniques that incorporates a novel data acquisition algorithm [7] and high accuracy analysis techniques to provide fast, quantitative 3-dimensional measurements of surfaces. The instrument can measure discontinuous features up to 100μ , scans a 10 micron range in 5 seconds, has an absolute accuracy specification of 1% and a smooth surface repeatability of 0.5nm.

SWLI profilers typically acquire and analyze the interferogram on-the-fly [8,9]. Computational bandwidth limitations result in slow acquisition, increasing the instruments susceptibility to environmental noise sources and limiting the resolution. To eliminate this bottleneck we have developed an efficient data acquisition algorithm that selects and saves only the narrow region of interference for later processing. For example, less than 4.3% of a 100μ scan needs to be analysed to produce the profile. The result is the ability to acquire data at video rates with substantial reductions in analysis time. Furthermore, higher density cameras can now be employed, improving the lateral resolution of the measurement with less of a speed penalty.

The data is well suited for an analysis using methods borrowed from Fourier-transform spectroscopy[10]. The data is analyzed in the frequency domain by mathematically breaking down the white light into separate colors, determining the interferometric phases of these colors and calculating the profile from these phases using

$$z = d\phi/d\omega,$$

which relates the optical path difference z to the rate of change of phase ϕ with color (as given by the optical frequency ω). The analysis produces exceptionally repeatable results without the need to revert to phase shifting techniques.

Typical sources of error found in SWLI profilers have been greatly reduced. The acquired interferogram is used to calculate the spectrum seen at the imaging sensor to correct for spectrum variations caused either by the illumination source or the test surface. Several distinct scan lengths are provided that are precalibrated with a distance measuring interferometer to eliminate scan nonlinearities. The reduction of these error sources are important steps in achieving the 1% accuracy specification.

Figure 1 is a profile from a measurement of a 20.37μ step-height standard using the SWLI profiler. The acquisition took only 11 seconds. The mean step height measured

was $20.25 \pm 0.02\mu$, in good agreement with the quoted step and a stylus measurement of 20.27μ . The long term repeatability of the measured step height was 20nm rms (0.1%) independent of source illumination or scan length.

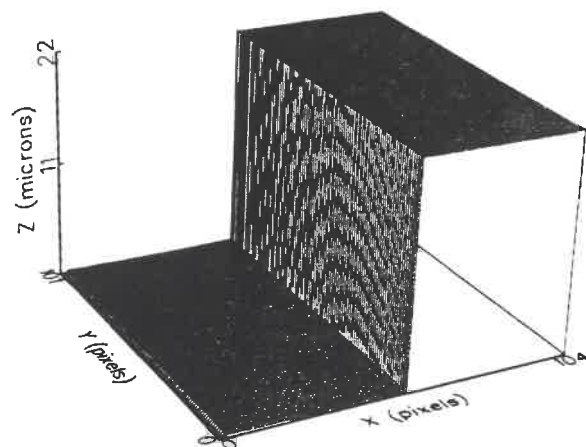


Figure 1: A 20.37μ step standard

Rough parts can also be accommodated as illustrated in Figure 2, which is a measurement of the surface of a stainless steel plate turned on a lathe. It has an RMS roughness of 832nm and the turning marks are clearly seen.

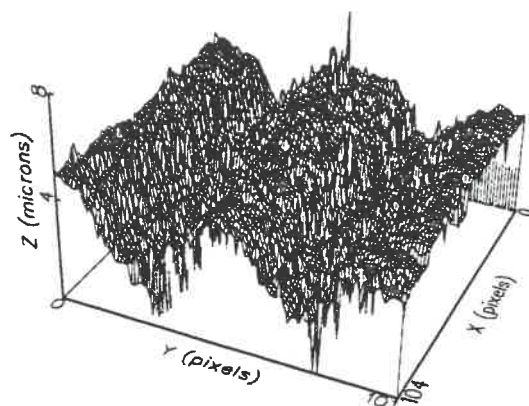


Figure 2: A stainless steel surface turned on a lathe

The instrumental smooth surface repeatability is 0.5nm as determined by the difference between two successive measurements of a smooth flat mirror under good environmental conditions.

In conclusion, we have developed a high speed, high accuracy profiler using scanning white light techniques. Both acquisition and analysis times are substantially reduced over conventional SWLI methods and compare favorably with PSI techniques. The data acquired is ideally suited for analysis using Fourier transform methods and the resulting profiles enjoy sub-nanometer vertical resolution.

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EVALUATION OF THE EFFECT OF ELASTIC RADIAL DEFLECTION DUE TO CENTRIPETAL ACCELERATION ON THE FINISH AND GEOMETRY OF PRECISION TURNED AND BORED COMPONENTS

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INTRODUCTION

Proper part location and clamping are critical factors in precision machining. While these elements are important to any machining operation, the range of tolerances for optimum parameters is much more narrow in precision operations than it is in more conventional procedures. As the tolerances and finish requirements become more demanding, successful production of a part also becomes more sensitive to the dynamic effects of the machining operation.

For example, in single point diamond facing of optical surfaces, spindle growth, a function of the square of spindle speed, has been shown to be a significant contributor to errors in finished surfaces [1]. Most of these surfaces are produced using spindle speeds that do not exceed 1000 RPM. These optical surfaces usually do not have tolerances and finishes on their diameters that are as demanding as those of the faced surface.

However, the increased performance and wear resistance of precision contact elements will demand that the machining industry begin to implement machine tools with the resolution and repeatability required to produce components in the sub-micrometer range [2]. As the number of types of parts being precision machined increases, so will the number of tolerances and machining parameters that are significantly affected by the dynamics of the machining operation. Since the cost of manufacturing the part is time dependent, the maximum cutting speeds possible will be used, which will also result in maximum dynamic effects.

The purpose of the work described here is to model one of these dynamic effects that is a function of turning speed, elastic radial deflection, and its subsequent effect on the radial depth of cut in a turning or boring operation. Possible effects on the finished part geometry and finish are discussed.

METHODS

Part to be Modelled

A bronze bearing that was diamond bored using a tool steel air bearing fixture was modelled for this work. Figure 1 illustrates the setup for the actual application. The bronze bearing has two counterbores, with diameters equal to .7550 inches, and a smaller axial through hole of .4606 inches. All three inner diameters were diamond turned.

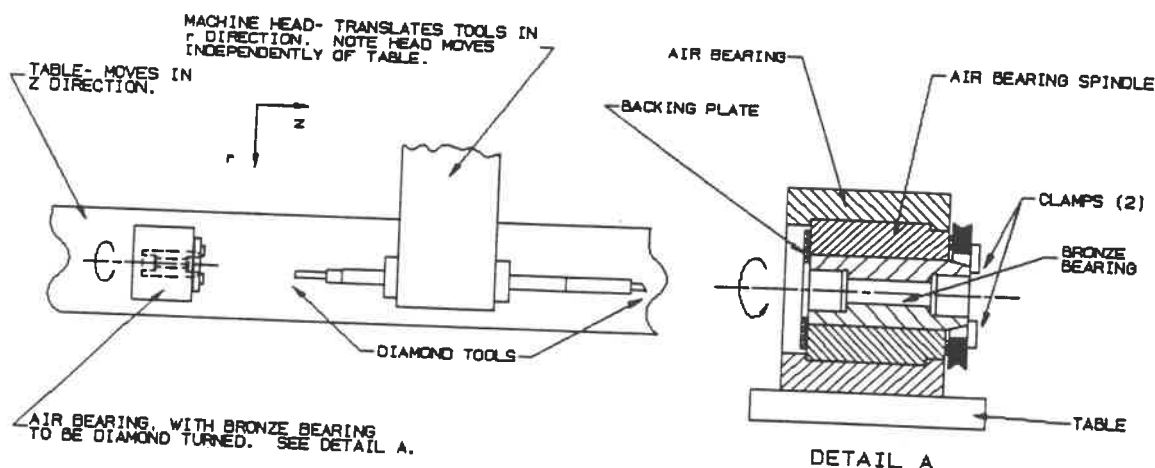


FIGURE 1: DIAMOND BORING SET UP FOR FINITE ELEMENT MODEL

The outer diameter of the bronze bearing was located in the air bearing with less than .0002 inches clearance, and also located against the backing plate. Two clamps maintained the position of the bronze bearing for diamond boring. This set up is described in detail in previous work by the author [3].

Analytical Relationship

The analytical calculation for radial deflection, δ , due to centripetal acceleration is given below [4]:

$$\delta = \left[\frac{\rho \omega^2}{gE} \right] \left[\frac{r}{8} \right] [3 + \mu] [1 - \mu] D$$

where

ρ = material density
 g = acceleration due to gravity
 μ = Poisson's ratio

ω = angular velocity of bearing
 E = elastic modulus of bronze
 r = radial location evaluated

and

$$D = b^2 + a^2 + \left[\frac{1 + \mu}{1 - \mu} \right] \left[b^2 \frac{a^2}{r^2} \right] - \left[\frac{1 + \mu}{3 + \mu} \right] r^2$$

where, b = outer radius of part, and a = inner radius of part.

Finite Element Model

The bronze bearing was modelled by using one half of the cross section as shown in Figure 2. Finite element analysis was performed in conjunction with an analytical solution, which does not account for all aspects of the problem. The finite elements were three node, constant strain, and triangular shaped. Because torsional effects were neglected, the axisymmetry allowed the bearing to be easily modelled in two dimensions.

Forces were applied to the radial degrees of freedom of all nodes. The forces are a function of material density, radial location, and angular velocity, as illustrated in the equation for radial deflection.

There were 732 triangular elements modelled using 429 nodes. All elements were identical in size, which was possible by eliminating some of the less critical geometry of the part. The analysis discussed here covers the two counterbores and inner diameter. Figure 2 illustrates the numbers of some of the nodes for reference. It can be observed that the numbers increase from left to right on Figure 2 (axially), and from bottom to top (radially). Element numbering is similar to nodal numbering.

Both axial and radial degrees of freedom on the outer diameter were constrained to zero displacement. When the presence of clamping effects was modelled, nodes 56, 110, 111, and 165 were also constrained to zero displacement.

Variables

Two variables were used in the analysis: RPM and presence or absence of clamping. Two values of RPM were used, 1000 and 4000. Although most ultra precision applications noted in the literature recommend speeds of 1000 RPM or below, the bearing modelled in this work was bored at 4000 RPM in order to manufacture the parts at a profitable rate [3].

The clamping effects variable was used to evaluate influences of local point loads.

RESULTS

Table 1 gives both minimum and maximum radial deflection at all four experimental levels for both the counterbores and .4606 diameter. There was no difference in radial deflection between the two counterbores.

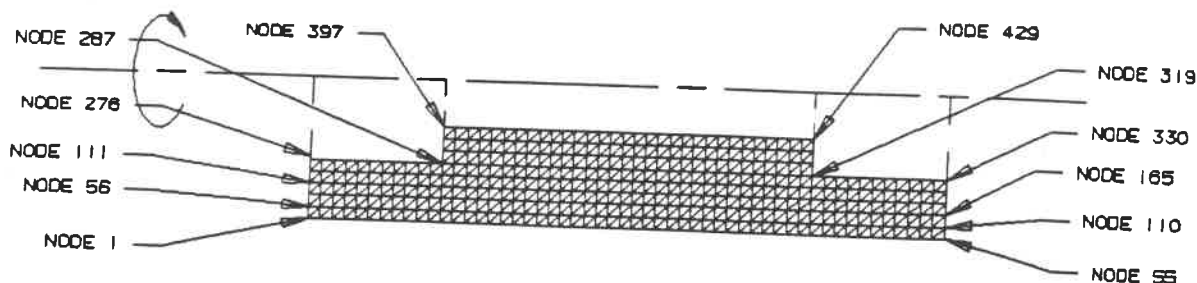


FIGURE 2: MESH OF BRONZE BEARING SECTION, USING 732 ELEMENTS, AND 429 NODES

TABLE 1: FINITE ELEMENT AND ANALYTICAL PREDICTED RESULTS.

Variables		Location in Bronze Bearing	Radial Deflection (micro inches)		
RPM	Clamps		FEM Min	FEM Max	Analytical
1000	Yes	Counterbore	1.06	1.62	N/A
		.4606 I.D.	2.42	2.55	N/A
	No	Counterbore	1.18	1.88	0.09
		.4606 I.D.	2.42	2.54	0.05
4000	Yes	Counterbore	17.0	25.9	N/A
		.4606 I.D.	38.7	40.7	N/A
	No	Counterbore	18.9	30.0	1.48
		.4606 I.D.	38.7	40.6	0.81

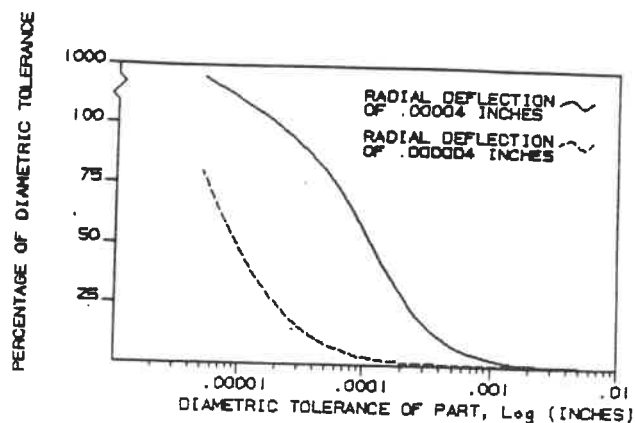


FIGURE 3: PERCENTAGE OF DIAMETRIC TOLERANCE TAKEN BY RADIAL DEFLECTION VERSUS DIAMETRIC TOLERANCE.

It can be observed that maximum deflection occurred in the smaller diameter at 4000 RPM. The higher value RPM resulted in a larger deflection in all cases, as the analytical relationship predicted. Table 1 also shows that models with clamping effects had a slightly larger radial deflection in all cases.

In general, the finite element model predicted a radial deflection that was an order of magnitude greater at the higher RPM in both the inner diameter and counterbores.

Effect of Part Geometry on Radial Deflection

Larger deflections were predicted in the smaller diameter than were predicted in the larger counterbores. Within the counterbore, however, there was a larger difference between minimum and maximum values than was predicted in the .4606 I.D.

For all experimental variable levels, maximum deflection was predicted at the ends of the .4606 I.D., which are nodes 397 and 429 on Figure 2. The minimum deflection occurred at the nodes located midway through this smaller diameter.

Maximum predicted radial deflection for the counterbore occurred at nodes 287 and 319, which are at the bottom of the counterbores. The radial deflection decreased along the counterbore until the ends of the counterbores, nodes 276 and 330, where deflection once again slightly increased. This relationship was consistent and independent of the clamping variable.

DISCUSSION

Variation of Deflection in Bearing

The maximum radial deflections are due to the effect of the bearing geometry on both loading and internal constraint to elastic deflection. Nodes 397 and 429, locations of maximum radial deflection for both the small I.D. and the entire bearing, have minimal constraints, since they are only shared by

two elements each. Therefore, there is minimal internal constraint to elastic deflection in both radial and axial directions [4]. This is also indicated by minimal radial and axial stress in these areas. The stresses are relieved by the relatively large deflections in these areas.

Nodes 276 and 330 also have minimal internal constraints to deflection, but they are not the sites for maximum radial deflection for the counterbores. The nodes at the bottom of the counterbores, 287 and 319, are the locations for the largest counterbore radial deflections. These results are due to the loads along the bottom of the counterbores being shared by a minimum number of elements.

Therefore, the loads due to centripetal acceleration are concentrated among fewer elements in that region, contributing to larger deflections. Using superposition of radial forces, it can also be observed that there are more nodal forces per unit length along the bearing axis between the two counterbores. This results in the larger radial deflections in the .4606 I.D. than occurred in the counterbores. As Table 1 illustrated, the clamps on the face of the bearing restrain material deflection in the axial direction, causing the stress to be relieved by greater radial elastic deflection.

Fraction of Tolerance

Finish cuts in machining are usually the same order of magnitude as the finish tolerances. The results illustrate that at speeds normally used in production, the elastic radial deflection will increase to levels that are large fractions of the total diametric tolerance of precision components. This relationship is illustrated in Figure 3, which plots the percentage of the diametric tolerance taken by the elastic radial deflection against the diametric tolerance of the part. Two sample values of radial deflection were used for this illustration, .000004 and .00004 inches. Note that .00004 inches was the maximum value of radial deflection predicted in the finite element model used in this work.

Figure 3 shows that below .0001 inches of diametric tolerance, the radial deflection that occurs while machining becomes a significant fraction of the finished tolerance of the turned component.

Effect on Material Removal

The results indicate that at 4000 RPM, the speed used to manufacture the bearing that was modelled here, the effective, radial, depth of cut would vary along the length of the bearing. Subsequently, surface finish, and diametric and geometric tolerances of the finished part would be functions of deflection.

Any deviations from nominal geometry of the part that exist before the finish cut would amplify the dynamic effects modelled here. For example, variation in the part diameter along its length would probably remain after the finish cut, due to the elastic recovery of the surface being machined. This condition is a result of the elastic radial deflection being of the same order of magnitude as the radial depth of cut and, subsequently, the finish tolerances of the bearing.

CONCLUSIONS

Finite element modelling of a precision boring operation indicates that machining parameters are sensitive to the dynamics of the set up, and that the degree of sensitivity is proportional to the relative precision of the process. Machining programs and tool paths may require additional offset information to compensate for dynamic effects while machining. Analytical relationships do not allow effects such as clamping and variation of geometry with respect to the axis of the part to be modelled.

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Surface Topography Measurement With A Fringe-Field Sensor

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Surface topography measurement is important in many areas of engineering because of the effect surface topography has on the functional performance of a part. However, because of the harsh operating conditions in a factory, no practical technique has been proven adequate for 3-D surface topography measurement in a manufacturing environment. The ideal 3-D surface topography measurement system for use in a factory needs to measure a surface at a high speed, the sensor must be rugged to withstand the harsh operating conditions, it should be portable to allow its use on a variety of surfaces, and it should be low in cost. This study proposes and evaluates a 3-D surface topography measurement system based on linear spiral sampling with a fringe-field capacitive sensor that may lead to the development of such a portable 3-D surface topography measurement system.

In linear spiral sampling, the sensor moves along a spiral path and the surface height is measured at the same angular locations on each revolution. Linear spiral sampling offers two advantages over rectangular sampling for surface topography measurement. First, a spiral scan allows a surface to be sampled faster than is possible with a rectangular sampling pattern because there is no need to continually accelerate and decelerate a sensor during sampling. Second, there is no hysteresis in the drive mechanism from continually changing the scanning direction.

From a spiral profile, surface height values on a rectangular grid are calculated from the Linear Spiral Sampling Theorem (originally used in NMR imaging) (Yudilevich and Stark, 1987)

$$z(\rho, \phi) = \sum_j \sum_k z(\rho_{jk}, \phi_k) \Psi(\rho, \rho_{jk}) \sigma(\phi, \phi_k) \quad (1)$$

Where the collection of spiral samples are denoted by $z(\rho_{jk}, \phi_k)$, $z(\rho, \phi)$ is the surface height in polar coordinates, ρ and ϕ denote radial and angular displacements, and $\Psi(*, *)$, $\sigma(*, *)$ are the radial and azimuthal interpolation functions of the form

$$\Psi(\rho) = \text{sinc} [2R (\rho - n/2R)] \quad \sigma(\phi) = \frac{\sin (N/2) \phi}{N \sin(1/2)\phi} \quad (2)$$

N is the number of angular samples, R is the radius of the sampled area, and n is the index (0, +1, +2, ...).

The fringe-field sensor consists of a 25 μ m diameter stainless steel wire encased in a ceramic shell. The ceramic forms a skid that maintains the electrode tip at a constant height as it moves across the surface. The perpendicular orientation of the wire allows the distance from the bottom of the sensor to the surface to be measured by detecting the variation in the electric field generated between the surface and the wire electrode. A fringe-field sensor is used as

the sensing element because of its small size, rugged construction, fast response, its insensitivity to vibration, and its ability to be used as a high speed sensor.

Sampling with a wire sensor represents a convolution of the actual surface with a 2-D spatial filter. Therefore, for some applications requiring high lateral spatial resolution, an estimate of the original surface is obtained by applying an inverse spatial filter to the measured surface.

An estimate of the sensor's spatial filter is obtained from a numerical model of the potential field between a wire electrode and a ground plane. The charge distribution in the ground plane is calculated using the method of moments (Hoole, 1989) and then the spatial filter is calculated from a knowledge of the charge distribution (Downs, 1993).

The static calibration and dynamic compensation for a sensor are calculated from the sensor's transfer function over a uniform lapped surface. The form of the transfer function is (Downs, 1993)

$$S(\omega_1, \omega_2) = A \exp(-2\pi B \sqrt{\omega_1^2 + \omega_2^2}) \quad (3)$$

Where S is the frequency response of the sensor, ω_1 and ω_2 are the spatial frequency components in cycles/mm, and A and B are parameters to be determined by dynamic calibration.

The accuracy of a fringe-field sensor was assessed by comparing 3-D surface topography measurements made using a fringe-field profilometer with measurements made with a skidless stylus profilometer (a Federal Surfanalyzer System 4000). These measurements were obtained by using the same test fixture to rotate and translate a test specimen so that a stationary sensor moves over the surface in a spiral path.

For the example presented here, a rough flycut surface with a dominant wavelength of $450\mu\text{m}$ and a straight profile R_a of $2.7\mu\text{m}$ was used as a test specimen. Five separate measurements were made of the flycut surface with each profilometer. For each test, the surface was measured using a spiral scan with 200 revolutions over a circular area with a radius of 3mm. The spacing between spirals was $15\mu\text{m}$, there were 2000 points taken at the same angular position on each revolution, and the rotational speed of the sample was 0.05 rev/sec.

The surface topography was reconstructed onto a square grid from the spiral profile using the Linear Spiral Sampling Theorem. There were 267×267 elements in the grid with a grid spacing of $15\mu\text{m}$. The reconstructed surface was leveled and a high pass cutoff filter was applied to the surface. For those surfaces measured with the fringe-field sensor, an inverse spatial filter was applied to the reconstructed surface.

Figure 1 and 2 show typical contour plots of the flycut surface measured with the fringe-field and stylus profilometers respectively. The contour plot is over a 4mm by 4mm area. The surface height values are gray scale coded from black equal to $-10\mu\text{m}$ ranging up to white equaling $+10\mu\text{m}$.

Comparing R_a values from each profilometer at the 5% significance level, an F -test showed no significant variation in measuring R_a between the two profilometers and a t -test showed that the two sets of samples appear to have equal R_a values.

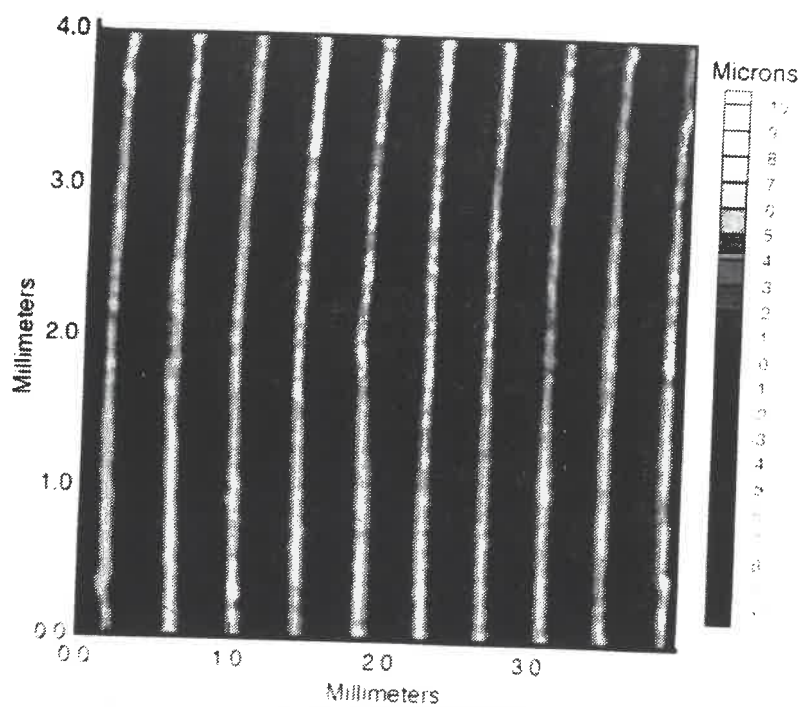


Figure 1 Contour Plot - Fringe-Field Profilometer

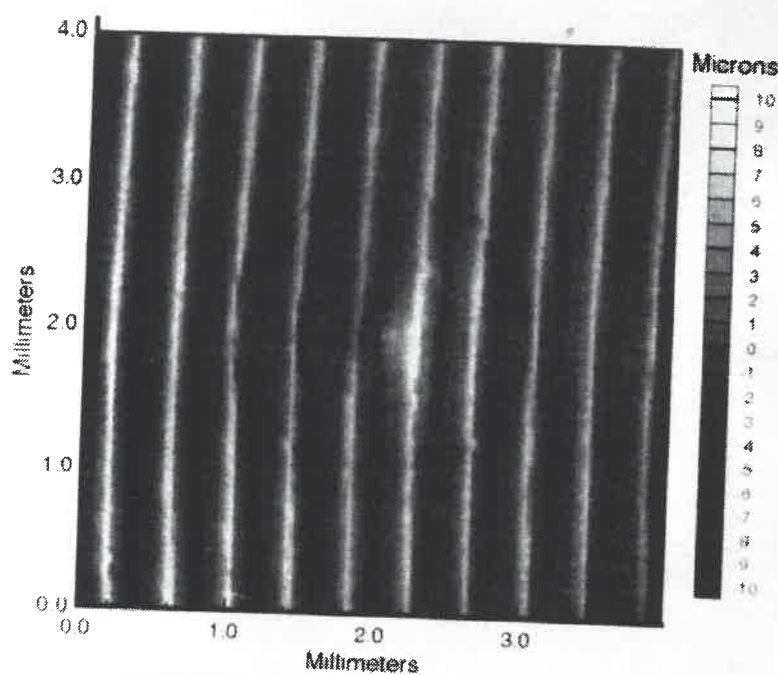


Figure 2 Contour Plot - Stylus Profilometer

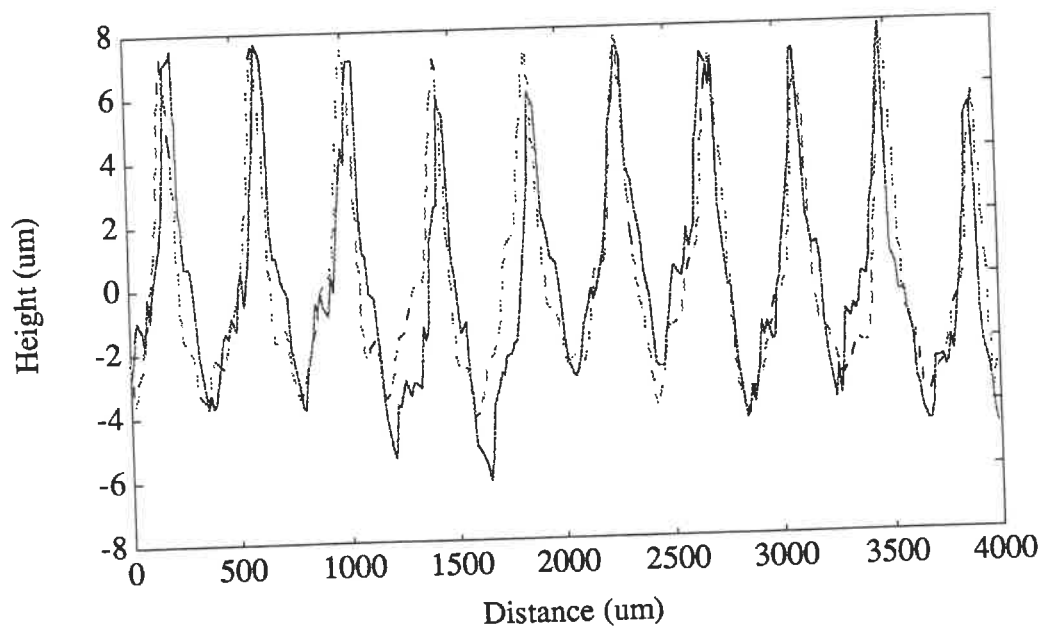


Figure 3 Stylus and Fringe-Field Profile Through Grid

The ability of the fringe-field sensor to measure a surface is seen in Figure 3, which shows a profile made through a restored surface grid. The solid line shows a profile through a stylus measured surface and the dashed line shows a profile through a fringe-field measured surface. There is some difference between the stylus and fringe-field profile because of the difference in location of the two profiles and the fact that the fringe-field is a skidded instrument and the stylus is a skidless instrument. But overall there is a good reproduction of the surface features.

Fringe-field capacitive profilometry using linear spiral sampling shows clear advantages for in-process measurements when compared with traditional techniques. First, since this method does not depend on a mechanical stylus following the surface height variations, the scanning speed is not limited by inertial effects. Profiling speeds several orders of magnitude greater than stylus methods can be obtained. Second, linear spiral sampling provides an advantage over raster scan sampling since this method does not require the sensor to repeatedly start and stop for each traverse. This allows scanning speeds several orders of magnitude greater than is possible with raster scanning.

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A New Type of Optical Stylus with Optical Skid

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1. Introduction

Recently, on-machine or in-process measurement has been gaining importance due to increasing demands for improved accuracy and productivity [1]. Under such measurement conditions, it is required that sensors not only have a long working distance, fast response characteristics, compact size and measurement datum not affected by the mechanical errors of machine guides, but also that they be resistant to environmental vibration and thermal drift. For surface roughness measurement, a long working distance and fast response characteristics can be obtained by using optical stylus method [2]. For the remaining requirements, the stylus method with a skid is thought to be a promising method.

Kiyono *et al.* have previously proposed an optical-skid surface roughness measurement sensor based on the astigmatic focus error detection method. The sensor was useful in reducing drift due to thermal deformations and noise due to environmental vibrations. However, the performance of the sensor as a surface roughness measurement device was not fully acceptable, in part because the optical system was too complicated, which created some alignment problems [3]. The focus error detection method using a linear beam and two photodiode arrays, as proposed by Kohno *et al.* [4], can also be used as an optical skid. However, the device is still in the developmental stages.

The most important characteristic of the optical skid is its smoothing effect on surface profiles. Huang *et al.* previously investigated this smoothing effect using the wide range which is a detecting range far away from the focal point in the critical angle method [5]. The experimental results showed a smoothing effect of approximately 1/3 of that obtained by a simple calculation using the spot area as the weighting factor. In this paper, a new optical-skid surface roughness measurement sensor is proposed using this wide range characteristic.

2. Principle and Configuration

All optical skid methods for measuring surface roughness use more than two beam-receiving optics to gather information from the optical stylus and optical skid. In this type of configuration, the optics of the sensor usually becomes complicated,

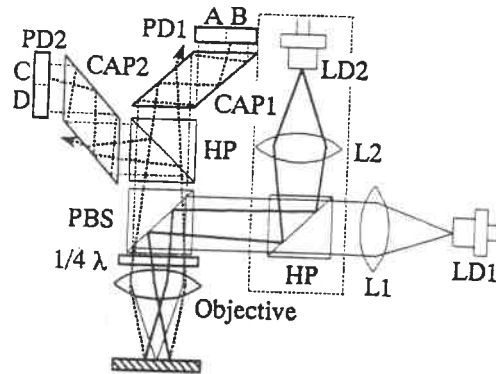


Fig.1 Optical system of the sensor

which not only makes the alignment of the optics difficult, but also makes the sensor large.

In this study, we focus on the fact that there are two measurement ranges, the precision range and the wide range, in the critical angle method and use two beams with different convergence angles to obtain the two measurement ranges simultaneously.

In the critical angle method, the measurement range near the position where the laser beam is focused on the specimen by the objective is called the precision range, and is usually used for displacement or surface profile measurement. Approximately 100 μm away from the precision range, there is another range within which the output of the sensor changes again with the displacement. This measurement range is called the wide range. In the prototype sensor described in this paper, two laser beams are generated using two laser diodes. One forms the precision range as the optical stylus and the other forms the wide range as the optical skid. This combination of optical stylus and optical skid is used for surface roughness measurement.

Figure 1 shows the configuration of the prototype sensor. The configuration is the same as that of the well-known optical stylus sensor based on the principle of the critical angle method, except for the light-source unit enclosed by dash lines. Laser diode LD1 is for the optical stylus and LD2 is for the optical skid. The optical skid beam is focused above the focal plane of the objective. At the focal plane, the spot size becomes quite large, with a diameter approximately 100 times that of the optical stylus spot. These two laser diodes are amplitude-modulated with frequencies of 20KHz

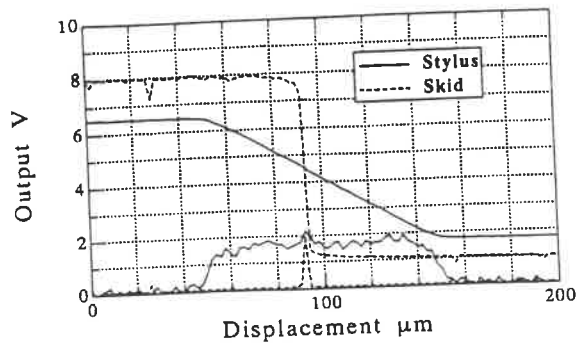


Fig.2 Spot diameters and distance between spots

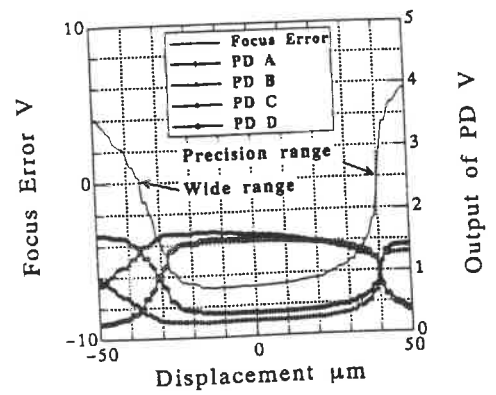
and 40 KHz respectively. The information obtained by the optical stylus and optical skid is separated by using the synchronous demodulation technique on the output signals from the photodiodes. In the following sections, the x axis is defined to be the horizontal plane in the paper, the z axis the vertical direction in the paper and the y axis the direction perpendicular to the paper.

3. Experimental Results

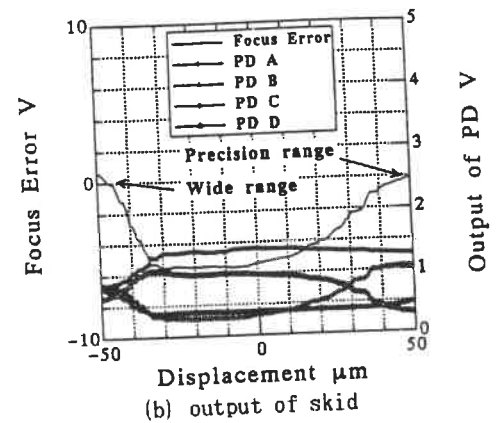
3.1. Characteristic Curves

Figure 2 shows the changes of the amount of light falling on the photodiode and the differential values for both the optical stylus and optical skid, obtained by scanning a piece of transparent glass with half of its surface coated with chromium through the laser beams. When the laser beams are projected on the coated surface, all the light is reflected back and the output from the photodiode is maximal. When the laser beams are projected on the edge of the coated surface, only some of the light is reflected back and the output from the photodiode decreases. When the laser beams are projected on the transparent glass surface, most of the light is transmitted through the glass and the output from the photodiode reaches a minimum. In Figure 2, the spot diameter of the optical skid is shown to be approximately 100 μm . Although it is not clear in this figure, another similar measurement with a higher lateral resolution found the spot diameter of the optical stylus to be approximately 1 μm . The deviation of the centers of the two beams along the x axis can be seen in the figure to be approximately 9 μm , and along the y axis approximately 7 μm .

Figure 3 (a) shows the output-displacement characteristic of the optical stylus. The horizontal axis is the displacement of the specimen along the z axis as monitored by a capacitance sensor. The precision range shown in this figure is used for the optical stylus. Figure 3 (b) shows the output-displacement characteristic of the optical skid. The wide range of this characteristic curve is used for the optical skid. Figure 3 also shows the outputs from the four photodiode elements. It can be



(a) output of stylus



(b) output of skid

Fig.3 Displacement-output relation

seen that they are approximately balanced.

3.2. Effects of Drift and Vibration

Figure 4 shows the thermal drift of the sensor over a time period of 30-minutes. The outputs of the optical stylus and optical skid fluctuate with the temperature with some delay, this fluctuation is mainly due to the change of the distance between the sensor and the specimen. However, this thermal drift is greatly suppressed in the differential output of the optical stylus and optical skid, as illustrated in the figure. Thus, the optical skid method is useful in greatly reducing thermal drift.

Figure 5 shows the short-term fluctuation of the sensor outputs. The noise due to environmental vibrations can be seen in the outputs of both the optical stylus and optical skid. However, the noise is greatly reduced in the differential output.

Figure 6 shows the effect of environmental vibrations caused by striking the table on which the experimental setup is fixed. Again the noise is almost completely canceled in the differential output.

3.3. Surface Profile Measurement

In the following three-dimensional results of surface profile measurement, (a) shows the results

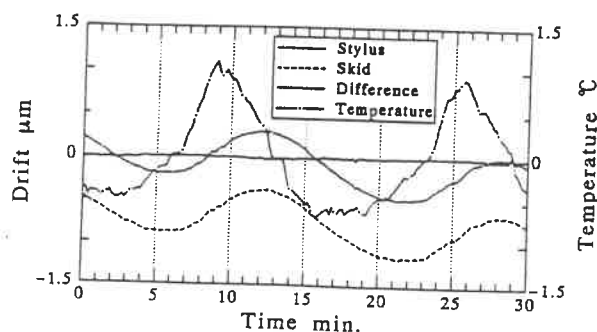


Fig.4 Thermal drift and cancellation

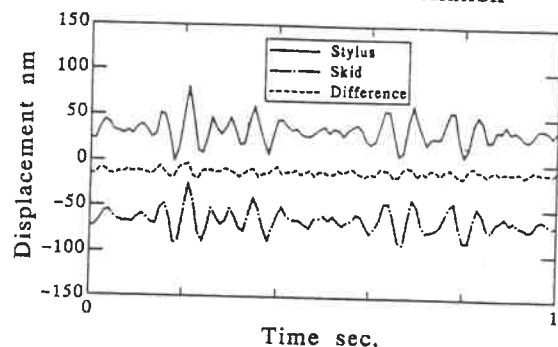


Fig.5 Circumferential disturbance

of the optical stylus, (b) shows the results of the optical skid, and (c) shows the differential results. Figure 7 shows the measurement results of a 300 line/mm reflection-type grating. The output from the optical skid is almost flat as a result of the smoothing effect. In a similar measurement of a 600 line/mm grating, a similar smoothing effect was observed. The measurement results of these gratings obtained by the optical stylus match both the design values and the measurement results obtained by STM and AFM. In the measurement of a 150 line/mm grating, the smoothing effect could still be observed, but the profile was only smoothed to 1/5 of its original height.

Figure 8 shows the measurement results of a milled surface. In the y-axis scanning of this measurement, there was a step-like scanning error, which can be seen in the results of both the optical stylus and optical skid. Because exactly the same error occurs in both the measurement results, the error is canceled in the differential output. It can also be seen that the surface profile measured by the optical skid is well smoothed.

Figure 9 shows the measurement results of a turned cylindrical surface with a diameter of 50 mm. For such large surface roughness, the smoothing effect of the optical skid is not necessarily perfect, but the differential result is not so different from the result of the optical stylus.

Figure 10 shows the results of a milled cylindrical surface with a diameter of 25 mm. Similar to the results of milled plane surfaces, the rough-

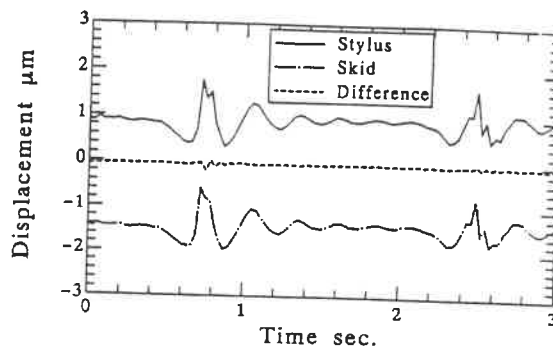


Fig.6 Forced impulsive vibration

ness of cylindrical surfaces can be effectively smoothed by the optical skid.

From the above results, it can be seen that even if the smoothing effect is not perfect when the surface roughness is large, the profile can still be reduced to 1/5 of its original height. When the machining accuracy increases, the accuracy of the datum generated by the optical skid also increases. Such performance of the optical skid is acceptable for applications in on-machine or in-process measurement.

4. Conclusions

The following results were obtained:

- 1) Using the optical skid, thermal drift and vibration noise can be reduced by at least a factor of ten.
- 2) Even for highly periodical surface profiles, such as gratings with 300 or more lines per millimeter (30 lines per diameter of the optical skid), the smoothing effect of the optical skid was found acceptable for real applications.
- 3) The smoothing effect of the optical skid for milled surfaces was also found acceptable for real on-machine surface roughness measurement.
- 4) Even for turned surfaces with very rough surface, the profiles can be smoothed to 1/5 of their original heights by the optical skid, which is acceptable for checking the surface roughness of work pieces on the machine or in the machining process.

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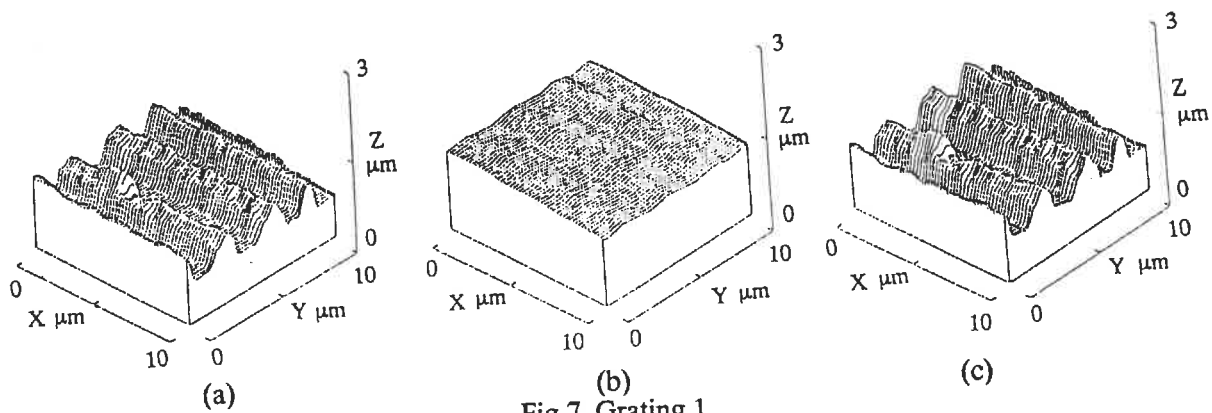


Fig.7 Grating 1

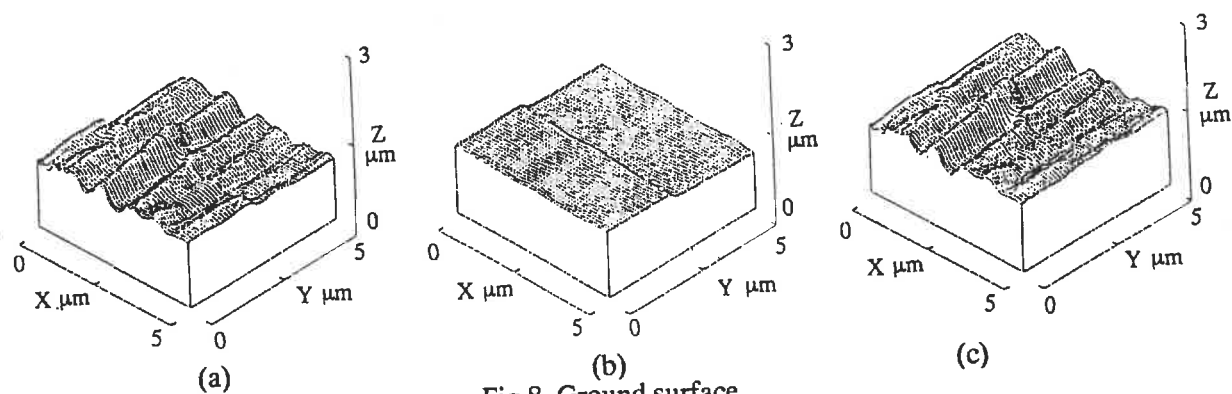


Fig.8 Ground surface

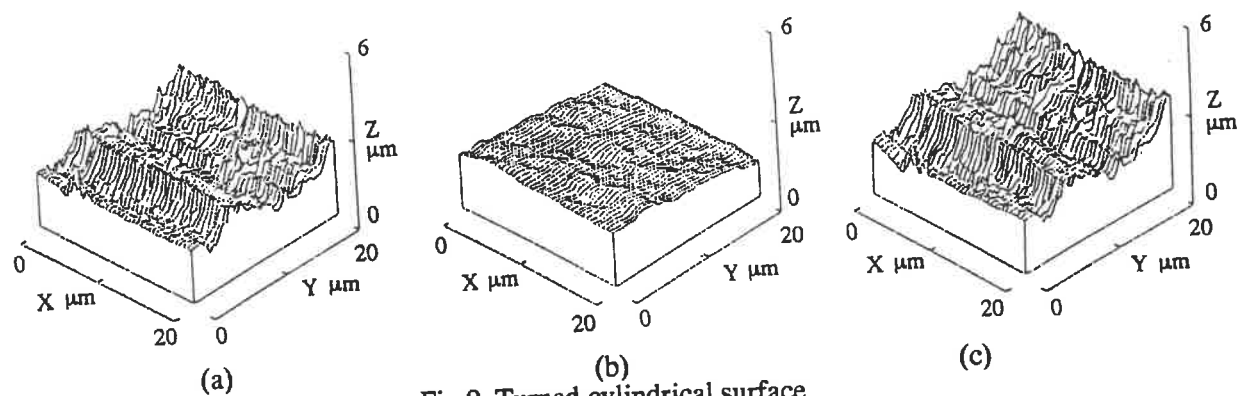


Fig.9 Turned cylindrical surface

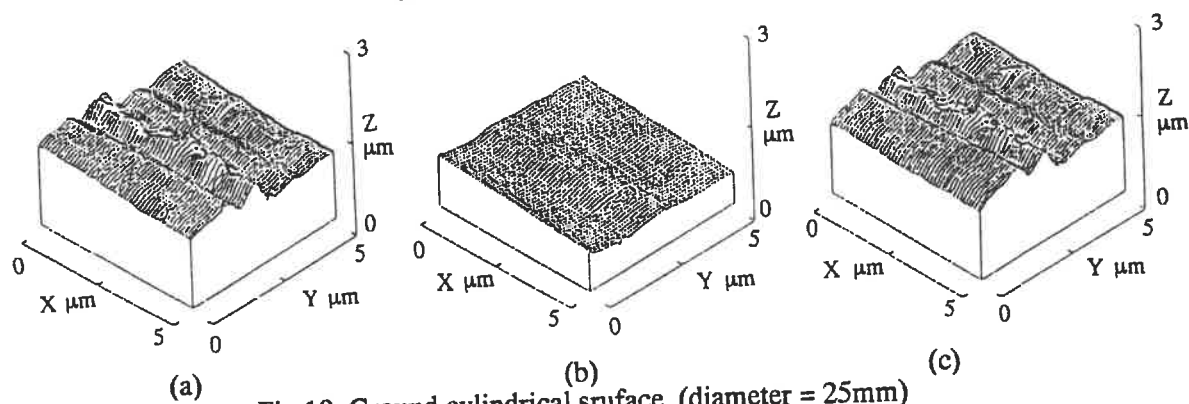


Fig.10 Ground cylindrical surface (diameter = 25mm)

TECHNIQUE AND RESULTS OF ANALYSIS OF SURFACES ORIENTATION CHARACTERISTICS ON VARIOUS SCALE LEVELS

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Purpose of the report is consideration of:

- (a) universal technique for the computer analysis of the surface orientation characteristic in response to digital images which are obtained with optical (OM), electron (SEM), and tunneling (STM) microscopes;
- (b) distinctions of it using on the example of number of technical and testing surfaces;
- (c) results investigation hard disks surfaces orientations characteristics.

TECHNIQUE

Based on using the certain method of images differentiation by application of the Sobel operator [1]. In the purpose of quantity estimation at the every image point azimuth angle of the brightness gradient is defined (for STM - height) and the histogram of frequency distribution onto twelve choosing directions in set up. In the graphic form the histogram is represented as a corresponding circular diagram which is symmetric about coordinate center.

The anisotropy coefficient $k \geq 1$ which is defined as ratio maximum frequency to minimum one and angle $\gamma \leq 90$ degree between these two main orientation directions have selected as the orientation integral characteristics. In case of the analysis of a samples with big on area deviations (defects, scratch, etc.) the further splitting of the frequency vector on the gradient levels (a frequency matrix and a set of circular diagrams) is used.

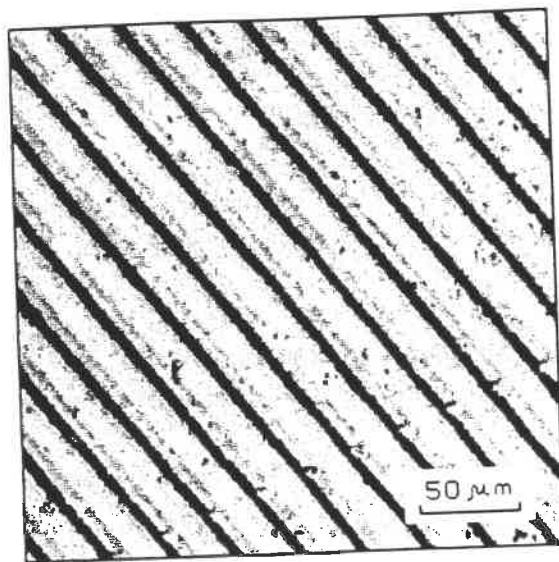
It is experimentally confirmed practical independence of the results from sample orientation in the field of vision and small sensitivity to a long waves distortions (non uniformity of illumination at OM, sample charging during scanning at SEM, sample slope at STM, etc.). Incorrect results can be obtained only with small (less then 40-50) dynamic range of a signal an image (aliasing effect).

INITIAL DATA

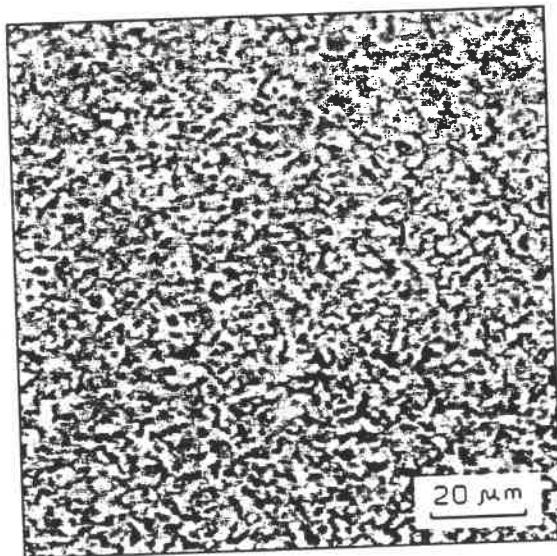
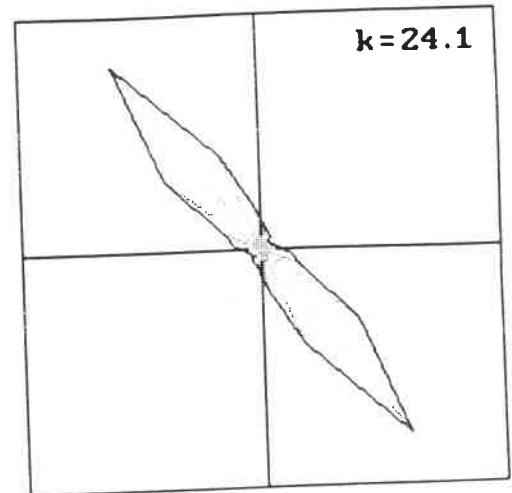
82 images of a different surfaces with the base length of scanning L from 0.1 to 1800 μ m obtained with OM, SEM, and STM [2].

LIMITING ANISOTROPY VALUES

The maximum value was estimated on the NIST testing sample with artificial unidirectional texture. For OM ($\times 200$) it was obtained $k=21$, for SEM ($\times 400$) obtained $k=24.1$ (Fig.1,a). For STM as a sample was used unidirectional surface of a diffraction



a



b

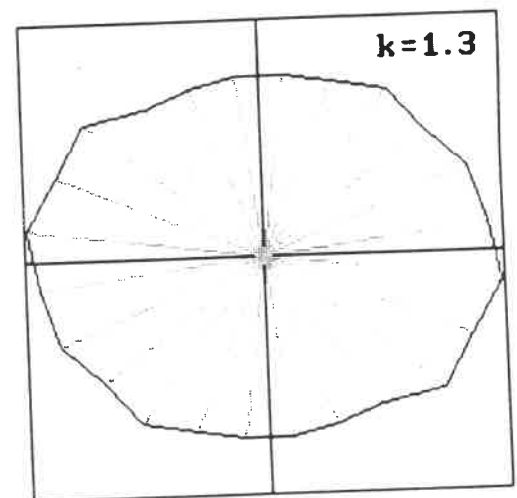


Fig.1. Limiting anisotropy values. a – strongly anisotropic surface, b – isotropic surface

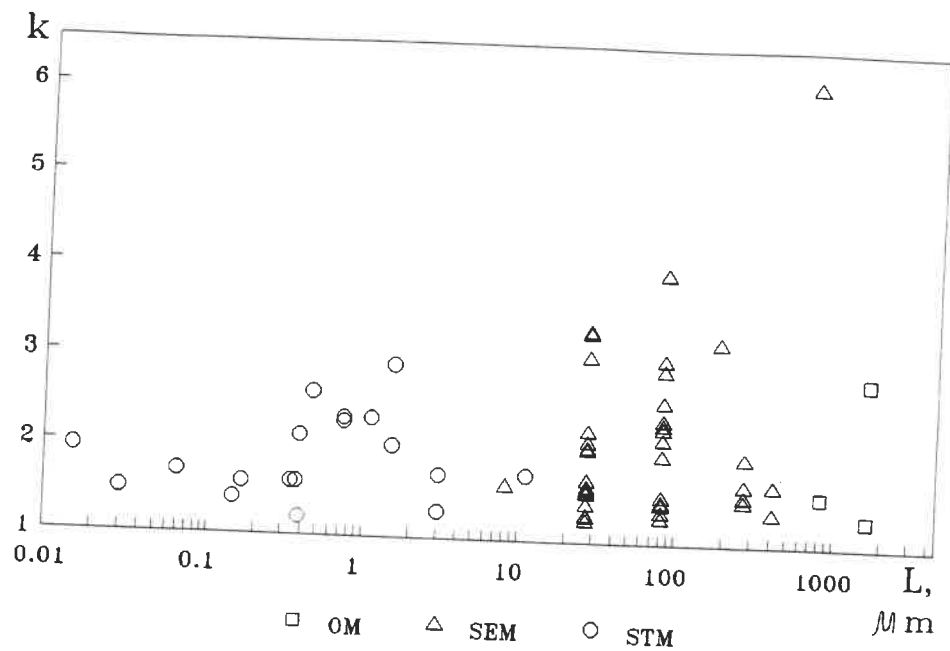


Fig.2. Dependence of coefficient k on scanning length of image L

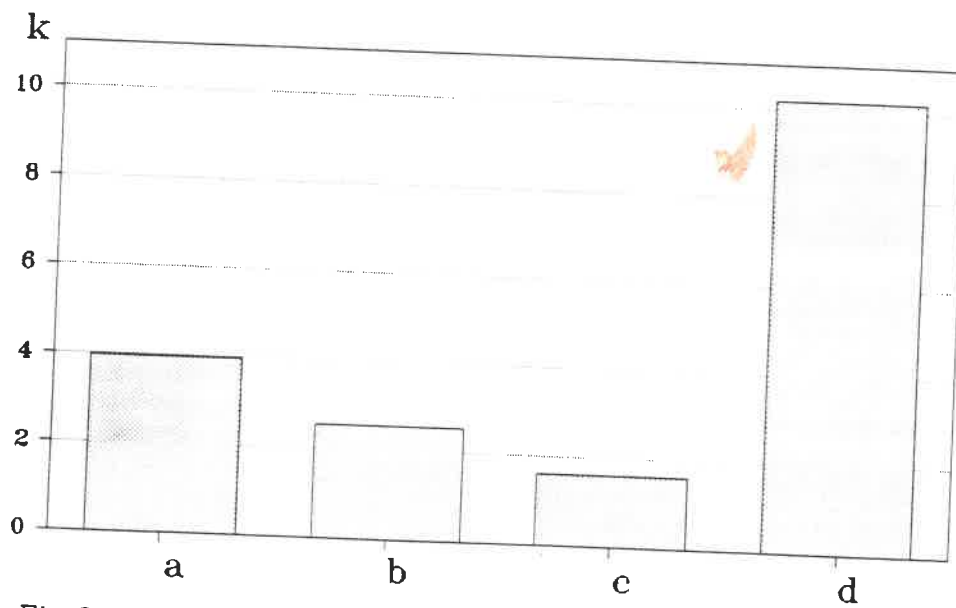


Fig.3. Variations in anisotropy coefficient for different stages of producing and operating of hard disk

lattice. With the scanning base $L=2.8 \mu\text{m}$ it was obtained $k=28$. The minimum values ($k=1.2-2.3$) were obtained for a magnetic disk base surface (Fig.1,b) processed by abrasive stream [2].

SCALE LEVEL AND ANISOTROPY

During investigation of the 61 images of real surfaces formed using various techniques of treating and operation processes the tendency for a increase of anisotropy range from $k=1.2-2.9$ with $L=0.1-10 \mu\text{m}$ to $k=3-6$ with $L=10-1000 \mu\text{m}$ is found (Fig.2). With the coefficient anisotropy increasing the angle between minimum and maximum anisotropy directions tends to 75-90 degree.

ORIENTATION CHARACTERISTICS TOLERANCE

It was investigated on the 9 SEM-images of the magnetic disks surfaces. With $\times 1000$ magnification for the three various regions anisotropy coefficients 2.9, 2.1, and 2.3 were obtained. With $\times 3000$ magnification the values respectively 2.0, 2.1, and 2.2 were obtained. All orientation diagrams had similar form.

ANISOTROPY CHANGE DURING DISKS PRODUCING AND OPERATING

Anisotropy coefficients are given on the diagram (Fig.3) for the surface at different stages of treatment. It was recorded, that in the course of hardening, which follows the substrate texture formation (a), anisotropy has decreased considerably (b). Such a result was anticipated in view of the plating effect of the additional surface layers(c). The same trend remains when magnetic and protective layers are deposited. Somewhat more unexpected, as it was mentioned earlier, is a very high anisotropy of the surface that has been subjected to friction (d). For example, after 10,000 start-and-stop operations executed by a magnetic disk-magnetic head system, there were high-frequency worn traces oriented towards the sliding direction, which made the surface highly anisotropic. After unidirectional machining of the magnetic disk substrate, the anisotropy coefficient was not very high.

Thus, more substantiated estimates of topography orientation were those done after completing of surface preparation. Variations in the orientational properties after operation should also be taken into account. Owing to them, even initially isotropic surfaces can acquire strong anisotropic structure. It's probably easy to find examples where anisotropy variations in the course of operation will have reversed trend (from strong anisotropy to isotropy).

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BEARING CURVE FOR ROUGH SURFACES IN QUASI-STATIC CONTACT - PHYSICAL MEASUREMENT METHOD

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As well known, the functional performance and durability of surfaces of machine parts in contact during exploitation process are critically conditioned by the phenomena occurring within the contact zone. They in turn depend on the state of surface layers.

The state of the surface layer, in general, depends on the physical features of the material and geometrical structure of the surface itself. To characterise the state of surface layer it is necessary to describe both these factors.

One of the first attempts to introduce such complex characteristic was the bearing curve (BC), proposed by Abbot and Firestone in the thirties. The bearing curve which is supposed to estimate dependence between approach (a) and real contact area (RCA), in fact describes only the geometrical features of the surface layer, thus its applicability is very limited. This situation has been changed by the possibility to conduct simultaneous measurements of both these parameters while the contact is subjected to pressure. The method described below provides means to register the (real) physical bearing curve (PBC) of the nominally flat surfaces under quasi-static pressure.

MEASUREMENT METHOD

The stand used for these tests was designed to provide means to examine the contact between the rough surface of the sample tested "a" with an "ideally" smooth and rigid surface of a reference sample "b" (Fig. 1).

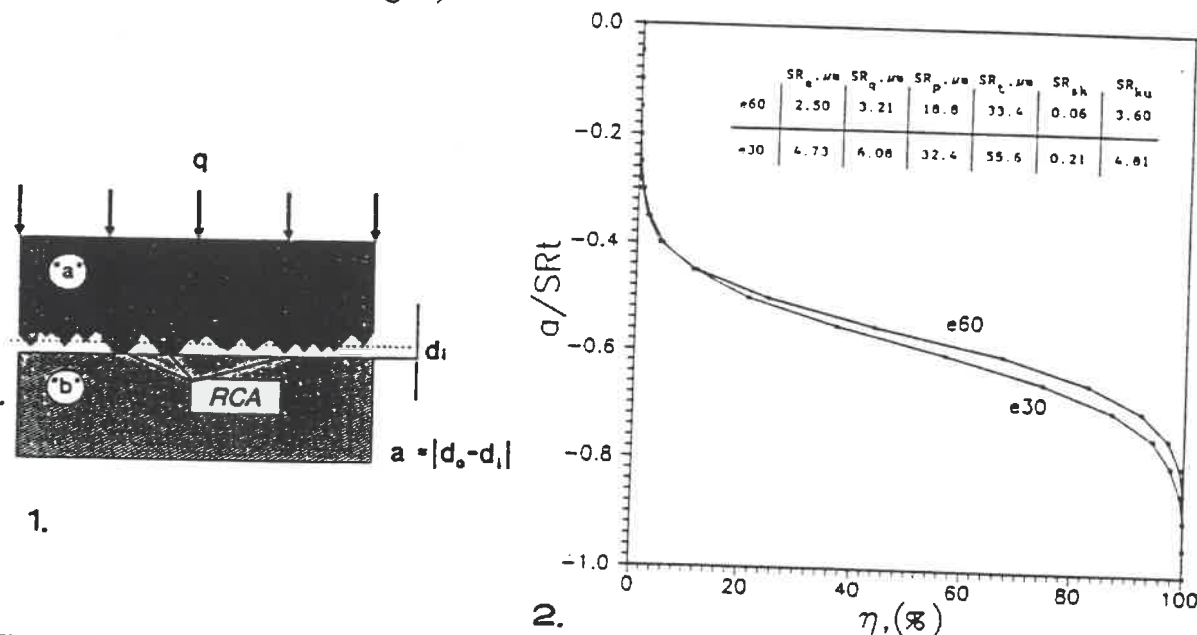


Figure 1. The configuration used for measurements of contact characteristics $RCA(q)$ and $a(q)$, (PBC).

d_0 - initial and d_i actual surface separation.

Figure 2. The bearing curves (BC) for surfaces of parts tested a/Rt - normalised height, η - bearing area ratio.

The configuration adopted is analogous to the one used for the determination of a "geometric" bearing curve (BC).

The simultaneous measurements of features $RCA(q)$ and $a(q)$, where q is the nominal pressure applied, have been conducted on a specialised instrument coupled with commercial appliances and a microcomputer. The stand, described in more detail in pos. [1, 2], utilised the following measurement methods and technologies:

- the echo method and the ultrasonic technology for RCA measurements;
- the Demkin method and the induction displacement measurement method for approach (a) evaluation;
- resistance strain tensometry method with the use of a force converter for the evaluation of applied pressure (q).

The instrument, located on the axis of a precise hydraulic press, was designed to provide a macroscopically uniform distribution of load over the contact plane, preceded by precision self-adjustment of the contacting surfaces. The approach measurements have been conducted by the core part elimination method, which provides the means for the measurements of surface microirregularities deformation [3] (a/Rt , where Rt is the maximum height of summit). The RCA was measured in terms of calibrated RCA was measured in terms of calibrated indications of the energy transmission ratio of a transverse ultrasound wave crossing the contact area.

The results of simultaneous $RCA(q)$ and $a/Rt(q)$ measurements have been used in the determination of the physical bearing curve (PBC).

MEASUREMENT RESULTS

The need to research the PBC can be justified basing on the following example. Let us take into consideration two steel samples (0.45% C) of different microgeometry, which display similar surface bearing curves (BC) as shown on Fig. 2. The surface layers of both samples have been similarly constituted, the only difference being the application of different alundum grain size in the final stage of abrasive blasting (samples e30 and e60).

The PBC curves for both surfaces subjected to loads reaching 300 MPa are shown on Fig. 3. The measurement errors are as follows: $q(\pm 3.5 \text{ MPa})$; $a(\pm 0.5 \mu\text{m})$; $RCA(\pm 2\%)$.

The comparison of the BC and PBC curves reveals different contact properties of both parts, not very visible in case of BC. A superficial analysis of the PBC curves proves that at the same a/Rt the e30 sample showed a much higher RCA than the e60 sample, and the acquirement of the same RCA is relatively connected with higher deformation of the e60 sample. Under stabilised load values the e60 sample displayed much stronger deformations showing a higher RCA, contrary to the results acquired for the e30 sample.

The PBC curves differ from the BC curves not only in shape, but also in orientation. The latter has been found particularly unstable in case of BC, relating the curve course to the highest roughness summit. Since even a negligently low matching load is able to deform the summits of this kind, any attempts to correlate the curves prove to be troublesome. Fig. 4 is an attempt to perform such a correlation. The BC curves have been shifted by the value of rp_k and $rp_k/2$, where rp_k is a parameter similar to the Sp_k (reduced summit height [4], corresponding to the negligence of deformation of the highest summits).

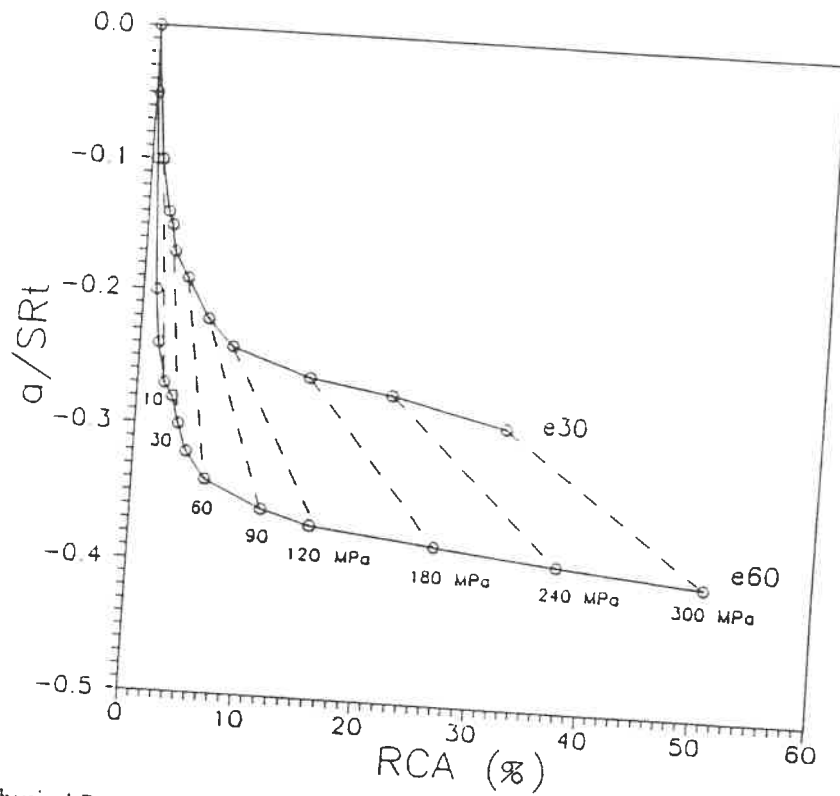


Figure 3. Physical Bearing Curves (PBC) of the behaviour of tested parts under applied pressure
 a/Rt - real contact strain; RCA - real contact area

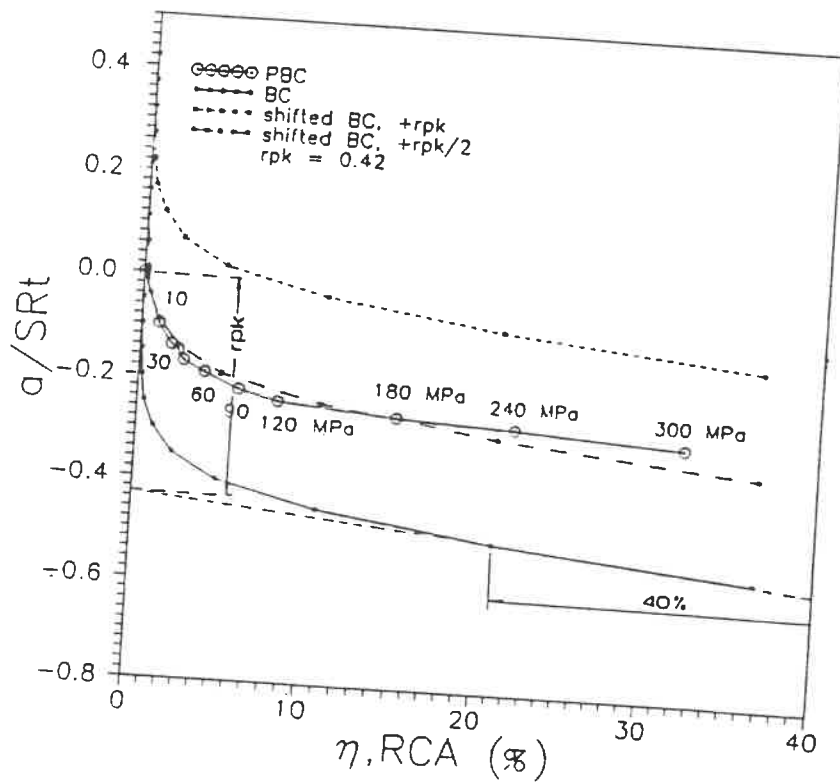


Figure 4. The comparison of location and shape of the PBC and BC curves.

CONCLUSIONS

The paper proves the feasibility of the research of physical bearing curves. The real surface bearing is usually different from the values expected basing on the Abbot-Firestone curve analysis.

ACKNOWLEDGEMENTS

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PARAMETERS FOR THE PREDICTION OF ROTATING BEARING PERFORMANCE

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INTRODUCTION

Roundness measurement, as it is known today, began with a need in the bearing industry for improved quality control. Metrology companies met those requirements by introducing measurement devices that employed precision rotary spindles and indicators for gathering data.

The rotary spindle provides a precision reference for the indicator. Data from the indicator, in the form of a graph, provides for the accurate assessment of the out-of-roundness of the part under test. By using various techniques, a numerical value can be applied to this out-of-roundness. This number represents the maximum peak to valley deviation from a true circle, and enables metrologists to perform meaningful inspections of round or cylindrical components, such as those used in bearing and piston manufacture.

This method of measurement led to a definite improvement in the quality of bearing components, and of other round and cylindrical parts. However, relying on this single number to describe out-of-roundness has limitations. For example, there may be ovality or tri-lobing in the bearing race. This will tend to mask smaller amplitude irregularities, which may be characterized as "bumps". In these cases, while the bearing is within its roundness specification, the irregularities can cause the bearing to chatter in service. This will lead to noisy operation and may result in excessive wear and shorter bearing life.

Over the years, several new methods were developed to detect the presence of such features and to determine their frequency and amplitude. Initially, the data was assessed using manual techniques.

Manual Techniques

One of the earliest methods of assessment was a visual technique, using an overlay. The basic requirement of this analysis is to determine the maximum peak-to-valley change over the width of the bearing contact area. This area is commonly known as a window.

An overlay, with a window of the required width drawn on it, is rotated around the chart and the worst case peak to valley within this window is determined. As with most manual methods of assessment, this is highly subjective. Despite this, much of the U.S. bearing industry has standardized on methods similar to this.

Advanced Techniques

No bearing company was satisfied with the accuracy or repeatability of manually assessing graphs. In the late 1970s computerized analysis systems were developed. This development process has recently culminated in DFTC (Departure From True Circle), Slope and Harmonic Analyses. The range of Rank Taylor Hobson Talyrond computerized gages include these three analyses.

DFTC Calculation

First, the operator should measure a properly centered part. The operator now needs to specify: the number of data points to be used in the analysis; the filter setting; the reference figure, that is Least square (LS), Minimum Zone (MZ), Minimum Circumscribed (MC) or Maximum Inscribed (MI); and the width of the window. The remaining steps in the calculation are then performed by the computer.

Step 1. The data is averaged down to the number of points selected. It is then filtered and fitted to the reference figure.

Step 2. The computer scans the window around the entire trace, rotating by one data point at a time. In this way, where for example 500 data points are selected, 500 windows are examined.

DFTC Results

Once data processing is complete, the result is displayed on the computer screen. The window where the greatest DFTC, ie the greatest change in peak-to-valley, was found is highlighted. This value is displayed with the out-of-roundness of the complete trace.

Slope Analysis

The premise of slope analysis is that the main problem with these bearings is more complex than simply high peak-to-valley changes over short distances. It is actually related to the slopes on the bearing surface. These slopes act as ramps that the balls must travel up and down as the bearing rotates. These will call the balls to constantly accelerate and decelerate, which causes the balls to collide, resulting in noise, wear and shorter bearing life.

Explanation of Slope Analysis

The following is a simplified explanation of how a Talyrond roundness gage performs slope analysis on a roundness trace. Slope is defined as the rate of change in radius with respect to angle, that is $\Delta R / \Delta \Theta$.

Again, the operator needs to select a filter and a reference figure, and to specify a window width. A roundness measurement is then made on a properly centered part, in the normal way. The computer then performs the following analysis steps.

Step 1. The data is averaged, filtered and the reference figure is fitted to the data.

Step 2. Each individual data point is assigned a slope value, as follows. For each data point, the three data points which precede it and the three which follow it, are assessed. A best fit line, which uses a special routine to weight each point according to its position relative to the center point, is fitted through these data points. The slope of this line relative to the best fit line for the entire trace is then determined. This slope value is then assigned to this particular data point. The computer then moves around by one data point, and repeats the process. In this way, each data point is assigned a slope value.

Step 3. The number of data points per window is calculated. To calculate the number of data points per degree, 360 degrees is divided by the total number of points used, then this is multiplied by the width of the window, which is specified in degrees. For example, using 500 data points and a five degree window would give: $360/500 = 0.72$.

In this example, there is one data point every 0.72 degrees, so a five degree window contains: $5 \times 1.4 = 7$ data points. Where the result is not a whole number of data points, the computer will round to the nearest whole number.

Step 4. The computer then examines each window through the entire roundness trace. The window rotates by one data point each time, so in this example 500 windows will be assessed.

Slope Results

Once computation is complete, the computer displays the maximum slope value and the average of all the slopes calculated. Units are either millionths of an inch, or microns, per degree.

Harmonic Analysis

In this process, data from a roundness trace is fed into a computer which performs a Fourier Transform on the data. Any repetitive data, such as a roundness trace, is made up of a series of events of different frequencies. The Fourier Transform breaks down the signal into its individual frequencies and then outputs, in either bar graph or tabulated form, the average amplitude of each individual frequency. This provides an in-depth analysis of the part, which is particularly applicable to process control and to the analysis of potential frequency effects when the part is in service.

Ovality Removal

Harmonic Analysis allows for a time saving feature to be included in the analysis software. The second harmonic of a roundness trace is ovality. If a part is insufficiently centered when it is measured, the effect is to show the part as oval, due to the changing stylus tip contact point. This effect can be removed from the result by removing the second harmonic and then recomputing the result. Use of this feature facilitates a saving in set up time. However, any actual ovality in the part will also be removed.

CONCLUSION

In conclusion, DFTC calculation, Slope Analysis and Harmonic Analysis are powerful tools which can help in determining the quality of a part and its suitability to its intended application. Recent advances in computer technology are increasing roundness measurement systems capability.

A STUDY ON THE CHARACTERISTICS OF EXTERNALLY PRESSURIZED CONICAL AIR BEARINGS WITH CIRCUMFERENTIAL GROOVES

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The externally pressurized air bearings are used as the main spindle bearings of the machine tools for small and light workpieces. Many researchers have worked for these bearings to have high stiffness and damping for more accurate runout of the spindle since 1970's. The Professional Instrument Co. have developed the axial groove compensated air bearing which had high stiffness. Kyosuke Ono[1,2] suggested the air bearing which had two circumferential grooves and discrete feeding hole restrictions. For this type of bearings, he concluded that it had higher stiffness, higher efficiency, and lower air flow than the grooveless bearings.

Generally, the externally pressurized bearings are composed of cylindrical journal bearing and annular thrust bearing in spindle system. However, the spindles supported by these bearings have the possibility of fluctuation due to the form error of the perpendicularity between annular thrust pad and cylindrical spindle. In order to overcome this problem, a conical bearing that supports both the radial and thrust load was used. The form error was reduced in conical shape bearing through maintaining high contact ratio between the spindle and bearing in manufacturing process. Moreover, since a unit of cylindrical journal bearing and thrust bearing can be replaced by one conical bearing, the total volume of the spindle system can be reduced.

We estimated the characteristics of the externally pressurized air journal bearing[3] efficiently using the direct numerical method[4]. We also developed a model for a grooveless conical bearing with 9 components of stiffness and damping coefficients compared with 4 components in cylindrical journal bearings[5]. In this paper, the direct numerical method is extended to estimate the characteristics of the conical air bearing with two circumferential grooves. The purpose of this paper is to investigate whether the stiffness and damping of the conical bearing can be increased by providing circumferential grooves as in the case of cylindrical bearings.

Fig. 1 shows schematic of the conical bearing system. In order to estimate the characteristics of the conical bearing system, the Reynolds' equation must be solved. The coordinate system for analysis is shown in Fig.2. The Reynolds' equation for a compressible fluid is as follows.

$$\frac{\partial}{\partial \theta} \left(PH^3 \frac{\partial P}{\partial \theta} \right) + \frac{\partial}{\partial \zeta} \left(PH^3 \frac{\partial P}{\partial \zeta} \right) = \Lambda(\zeta) \frac{\partial}{\partial \theta} (PH) + 2\Lambda_c \frac{\partial}{\partial \tau} (PH) \quad (1)$$

Assuming that the motion of the center of rotor shaft is very small in the vicinity of the steady state point, the perturbed Reynolds' equation is derived by perturbing the film thickness and the film pressure.

$$\nabla \cdot [P_o H_o^3 \nabla P_o - \Delta(\zeta)(P_o H_o)] = 0 \quad (2)$$

$$\nabla \cdot [P_o H_o^3 \nabla P_k + 3H_o^2 H_k P_o \nabla P_o + H_o^3 P_k \nabla P_o - \Delta(\zeta)(P_o H_k + P_k H_o)] = i2\Delta_c (P_o H_k + P_k H_o) \quad (3)$$

where $k = x, y, z$ and $H_o = 1 - \cos(\theta - \phi) \cdot \cos \beta$, $H_x = -\cos \beta \cos \theta$, $H_y = -\cos \beta \sin \theta$, $H_z = -\sin \beta$.

To consider the discrete change of film thickness at the groove boundaries, the film thickness function $H(i,j)$ at each node is composed of 4 components as shown in Fig.3. This is equivalent to a successive stepped film thickness.

The dimensionless mass flow rate through the inherently compensated air supply restrictor is obtained by assuming the isentropic process for compressible fluid. The perturbed mass flow rate equation can be obtained as in the perturbed Reynolds' equation.

$$Q_s = Q_o + Q_x \Delta x + Q_y \Delta y + Q_z \Delta z \quad (4)$$

The equations (2) and (3) are integrated over the lubricated surface Σ_{ij} which encompasses the arbitrary point (i,j) in Fig.4, and then the surface integrals are converted to line integrals by the Gauss' divergence theorem. The integrals result in the continuity equation for mass flow rate. At the mesh points located in air supply restrictors, the inlet mass flow rate should be added to the continuity equation. The functions of pressure are assumed to be linear in the finite control volume, and by using a 9-node finite difference equation the following algebraic equation is derived for each mesh point.

$$A_{oi,j} P_{oi,j} = A_{oi-1,j} P_{oi-1,j} + A_{oi+1,j} P_{oi+1,j} + A_{oi,j-1} P_{oi,j-1} + A_{oi,j+1} P_{oi,j+1} \\ + A_{oi-1,j-1} P_{oi-1,j-1} + A_{oi+1,j-1} P_{oi+1,j-1} + A_{oi+1,j+1} P_{oi+1,j+1} + A_{oi-1,j+1} P_{oi-1,j+1} + Q_o \quad (5)$$

$$A_{ki,j} P_{ki,j} = A_{ki-1,j} P_{ki-1,j} + A_{ki+1,j} P_{ki+1,j} + A_{ki,j-1} P_{ki,j-1} + A_{ki,j+1} P_{ki,j+1} \\ + A_{ki-1,j-1} P_{ki-1,j-1} + A_{ki+1,j-1} P_{ki+1,j-1} + A_{ki+1,j+1} P_{ki+1,j+1} + A_{ki-1,j+1} P_{ki-1,j+1} + Q_k \quad (6)$$

The equation (5) is nonlinear because the coefficients A_o 's have the pressure terms while the equation (6) is linear algebraic equation. To solve the equation (5), the iteration method is used. The boundary conditions to calculate P_o 's and P_k 's are; (1) the pressure is atmospheric at both ends of the bearing, (2) all perturbed pressures P_k 's are zero at both ends of the bearing, and (3) the pressure distribution is periodical.

The stiffness and damping coefficients can be obtained by integrating the solved P_k 's over the bearing surface.

$$K_{jk} = \frac{k_{jk} C_r}{P_a (r_i + r_o) L}, \quad j = x, y \quad k = x, y, z, \quad K_{zk} = \frac{k_{zk} C_r}{P_a \pi (r_o^2 - r_i^2)}, \quad k = x, y, z \\ D_{jk} = \frac{d_{jk} \omega C_r}{P_a (r_i + r_o) L}, \quad j = x, y \quad k = x, y, z, \quad D_{zk} = \frac{d_{zk} \omega C_r}{P_a \pi (r_o^2 - r_i^2)}, \quad k = x, y, z \quad (7)$$

The analytical method presented above is applied to a conical air bearing where the dimensions are $r_i = 32.5$ mm, $r_o = 70$ mm, $B = 80$ mm, and $C_r = 10$ μ m. There are 4 feed holes in the groove along the circumference and the distances between adjacent holes are assumed to be exactly same.

Fig.5 shows the influence of the groove depth b_g / C_r on the stiffness coefficients when compressibility number Λ is 1.0. There are 9 components of stiffness because the movement in one direction affects the stiffness of all three directions. The coupled stiffness coefficients $K_{yx}, K_{zx}, K_{xy}, K_{zy}, K_{xz}, K_{yz}$ are small compared with the principal ones K_{xx}, K_{yy} and K_{zz} . There exists a groove depth which provides a maximum stiffness for a given dimensions. The groove depth ratio at the maximum stiffness is about 2.0. This maximum stiffness K_{xx} is about 1.8 times higher than the stiffness of the grooveless bearing. Fig.6 shows the influence of the groove depth on the damping coefficients. The damping coefficients decrease with the increase of the groove depth. Unfortunately, the damping of the groove bearing is less than that of the grooveless bearing.

Fig.7 and Fig.8 show the principal stiffness and damping coefficients as a function of distance between two grooves for various widths of groove. The stiffness coefficients have the

maximum value when the distance ratio g_i/B is about 0.8. However the damping coefficients have the minimum value at the same condition.

Fig.9 and Fig 10 show the 9 components of stiffness and damping coefficients as a function of eccentricity ratio $\epsilon = e/C_r$.

In conclusion, the characteristics of the conical air bearing was successfully estimated by the direct numerical method. Analytical results show that the groove bearing has higher stiffness than grooveless bearing but has lower damping. Therefore it seems that the groove conical bearing can be well applied to the spindle of high precision machine tools.

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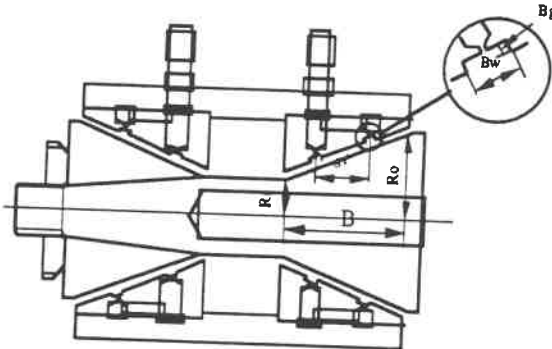


Fig.1 System configuration of conical bearing

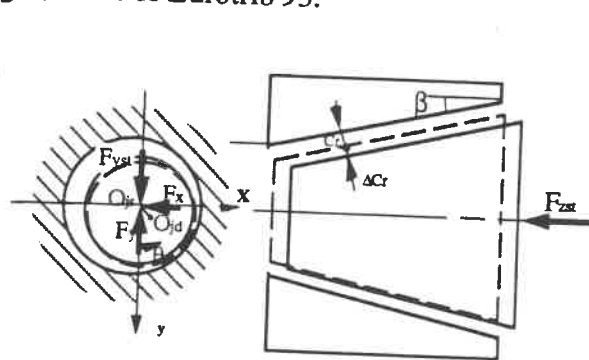


Fig.2 Coordinate system for analysis

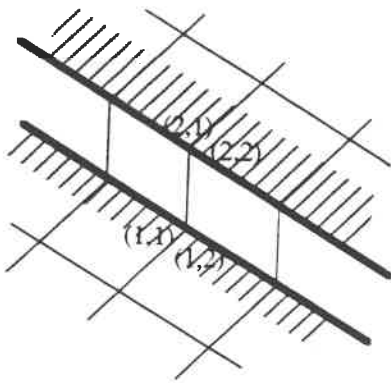


Fig.3 Film thickness at one node

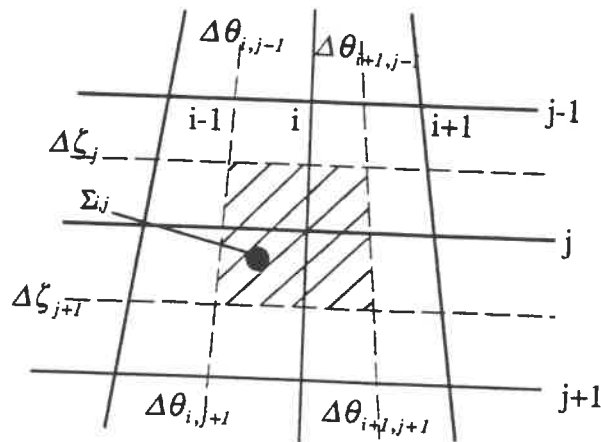


Fig.4 Mesh for analysis

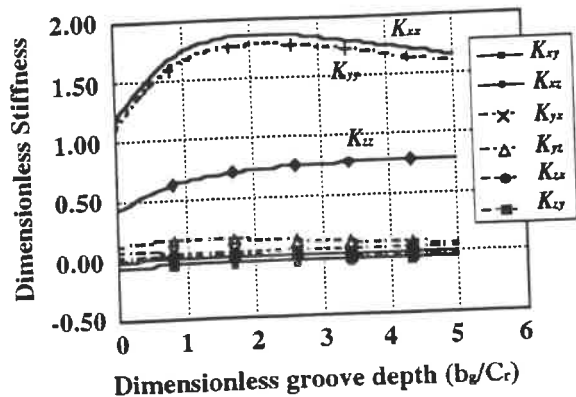


Fig.5 The influence of groove depth on the stiffness

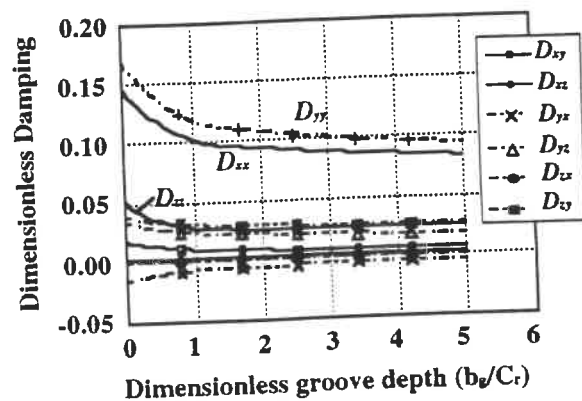


Fig.6 The influence of groove depth on the damping

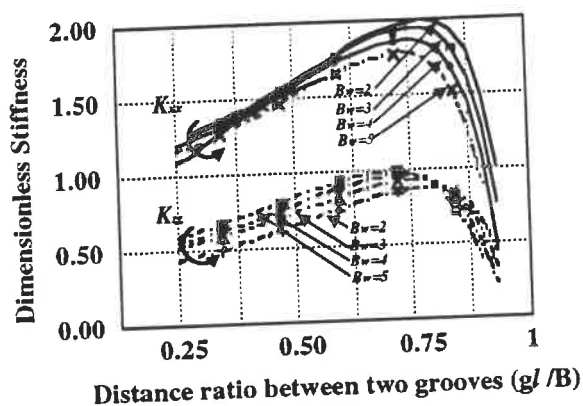


Fig.7 The influence of distance between two grooves on the stiffness

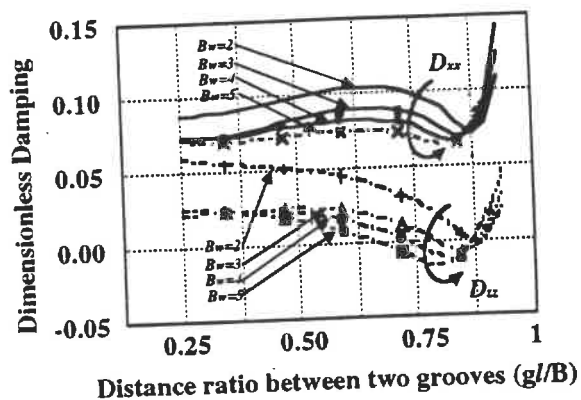


Fig.8 The influence of distance between two grooves on the damping

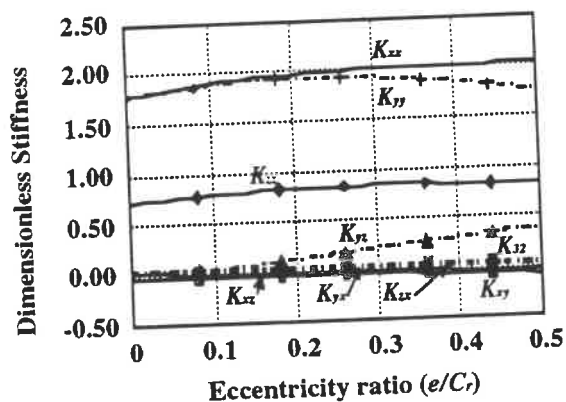


Fig.9 The influence of eccentricity ratio on the stiffness

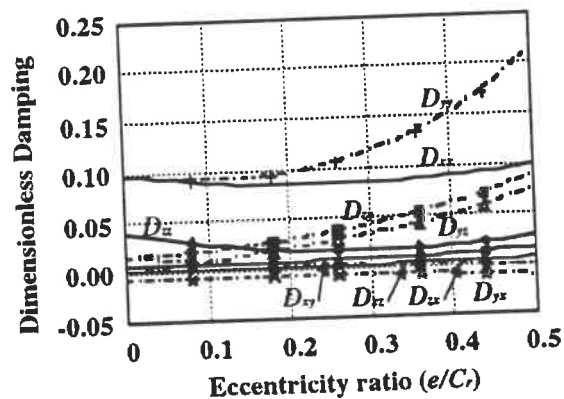


Fig.10 The influence of eccentricity ratio on the damping

DEVELOPMENT OF A VIBRATION GRIPPER

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1. Introduction

When a robot hand grasps an object, the robot must have data on the weight of the object and the coefficient of friction between the object and the robot finger for determining the most suitable grasping force. However we can not completely provide such data on all object to be grasped because there are many kinds of objects that a robot hand might be required to grasp. Therefore, when a robot hand grasps an object without available data, the robot must determine the most suitable grasping force by itself. The most suitable grasping force means that force by which the object does not slip down and the object is not damaged or distorted. Intelligent grippers that can determine suitable grasping forces automatically have been developed.^{1) 2)} These grippers determine the suitable grasping force by multiplying the distance which the object has slipped down by a certain constant. However the constant is determined by experiment; therefore, if the constant is not reasonable, the object will slip down or will be damaged. Moreover, if the weight of the object becomes light after the initial grasping process (for example, because of water being poured from a glass held by the robot hand), the grasping force could apply excessive force because the slip distance is not generated, and these grippers can not determine the suitable grasping force without the numerical value of the slip distance.

In this paper, we present a new vibration gripper which is able to handle an object of unknown weight and unknown surface conditions with its most suitable grasping force. Additionally, the gripper can adapt the grasping force to the most suitable value of the object even if weight and surface condition of the object changes.

2. Grasping method

The gripper is constructed to have two parallel fingers as shown in Fig.1. Each finger consists of a structural finger, a support spring, a grasping finger, an absorber spring, and an oscillator block. We call the grasping finger and the oscillator block a vibration finger and a damping oscillator in the right finger, and a receiving finger and a receiving oscillator in the left finger. These parts of the right finger and the left finger constitute series mass-spring systems having two degrees of freedom. In the vibration finger, external force p_0 is generated by rotating an eccentric mass m_P which radius is r . The

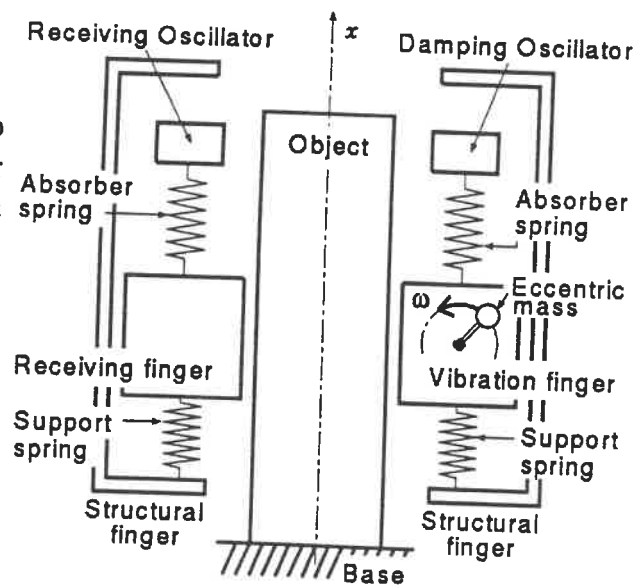


Fig.1 Structure of the vibration gripper

gripper holds an object by sandwiching between the vibration finger and the receiving finger.

In this gripper, the vibration finger is equal to the receiving finger in mass and its mass is represented by m as shown in Fig.2-1. The damping oscillator is equal to the receiving oscillator in mass and its mass is represented by m_o . The spring constants of the supporting spring and the absorber spring are represented by k and k_o .

We shall consider the case in which the object parameters for mass and surface condition between the object and the grasping finger are follows:

M is the mass of the object,

μ_s is the coefficient of static friction.

and we assume that a slip occurs between the object and the grasping finger when the force acting on the contact surface between the object and the grasping finger exceeds the maximum static friction force f_{MAX} . The value of f_{MAX} is depend on the grasping force F and its value is $\mu_s \cdot F$.

Through following processes, the object is grasped with the most suitable grasping force.

[Process 1 : (Fig.2-1)] Before the gripper grasps the object, the gripper regulates the number of revolution of the eccentric mass as the angular frequency ω of the external force is correspond to the natural angular frequency ω_o of the system composed by the absorber spring and the damping oscillator, so the amplitude of the vibration finger becomes zero³⁾ and damping oscillator vibrates with the amplitude A_o which value is $m_p r / m_o$.

[Process 2 : (Fig.2-2)] When the vibration finger and the receiving finger contact with the object and the gripper grasps the object with a little force, the external force is partly transmitted to the object. In Fig.2-2, the transmitted force is represented by p . Therefore, the amplitude of the damping oscillator becomes shorter than its amplitude in the process 1. In the process 2, the object has been put on the base. The receiving finger and the receiving oscillator remain stationary.

[Process 3 : (Fig.2-3)]

The gripper raises the structure fingers to a height of Δx in the process 3. When the raised height of the structure fingers is x , the spring force R which value is kx acts on the grasping finger and the receiving finger, respectively. Therefore, the combined force $R+p$ acts on the contact surface between the vibration finger and the object and the spring force R acts on the contact surface between the object and the receiving finger. Although the total force $2R+p$ acts on the object, the object does not rise up and the vibration finger slips up because the total force $2R+p$ for raising the object is smaller than the weight of the object and the force acting on the contact surface becomes bigger than the maximum static friction force f_{MAX} before the structure

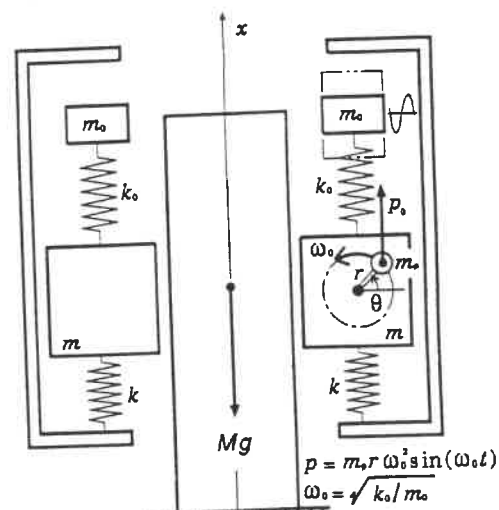


Fig.2-1 Process 1

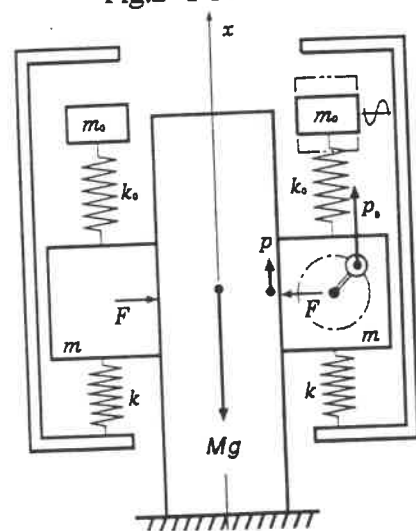


Fig.2-2 Process 2

Fig.2 Initial grasping process

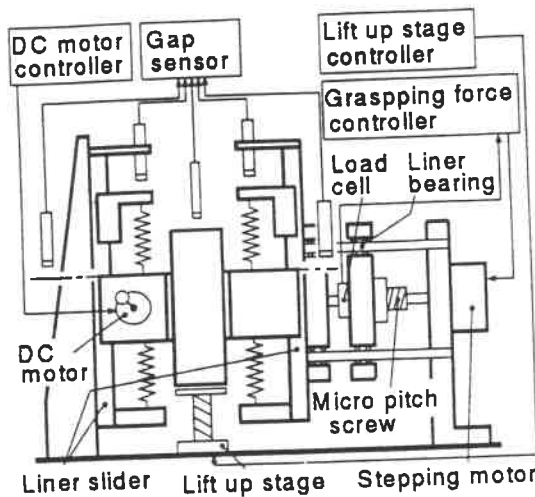


Fig.3 Experimental apparatus

following grasping force F_0 :

$$F_0 = \frac{Mg}{2\mu_s} + \frac{m_p r \omega_o^2}{2\mu_s}$$

This grasping force F_0 is composed of the first term (i.e. the necessary grasping force to support the weight of the object) and the second term (i.e. the necessary grasping force to be transmitted the half external force for vibrating the receiving oscillator). Therefore, the F_0 is almost correspond to the minimum grasping force for grasping the object without slipping because it is possible to make the small eccentric mass and the external force becomes small.

In the grasping condition completed the process 5, the grasping force becomes the most suitable grasping force F_0 and the gripper grasps the object with the most suitable grasping condition, that is, the vibration finger and the receiving finger are stationary while grasping the object and the damping oscillator and the receiving oscillator vibrate with the half amplitude of its in the process 1.

3. Experimental results

The experimental apparatus is shown in Fig.3. In the apparatus, grasping force was regulated by the stepping motor and the micro pitch screw and the object was lowered by

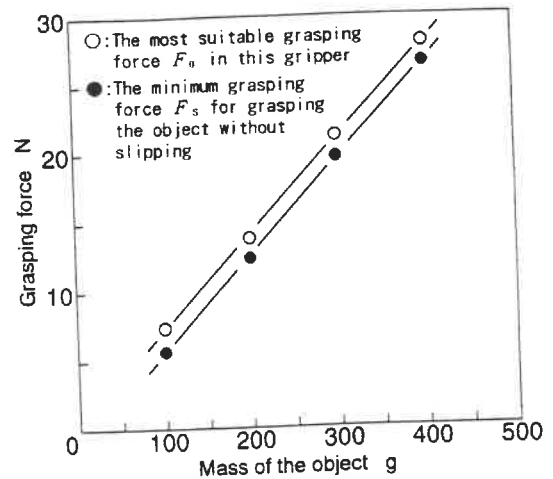


Fig.4 The most suitable grasping force

using the lift-up stage because the structural finger could not raise the object. Surface material of the object was Teflon and surface material of the vibration finger and the receiving finger was acrylic resin, and mass of the object was adjusted to 100g, 200g, 300g, 400g by making some holes on the core plate.

In the experiments, the most suitable grasping force F_0 was measured after the gripper accomplished the process 5 and the minimum grasping force F_s was measured without rotating the eccentric mass.

The values of F_0 and F_s were shown in Fig.4. It was made clear that the most suitable grasping force was almost correspond to the minimum grasping force.

4. Conclusions

- 1) The newly developed gripper can grasp objects with the most suitable grasping force F_0 by executing the five processes.
- 2) It was confirmed by the experiment that the most suitable grasping force F_0 is almost correspond to the minimum grasping force F_s .

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ON THE IMPROVEMENT OF THE DAMPING CAPACITY OF MACHINE TOOL STRUCTURES BY COMBINING WITH HIGH DAMPING MATERIALS

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Introduction

In order to achieve higher productivity and machining quality it is essential to suppress undesirable vibrations caused during machining process. For this purpose, filling a machine tool structure with a high damping material was experimented⁽¹⁾. However, the method increases the weight of structures and in some cases the damping capacity can not be improved enough. Therefore, more detailed studies are necessary to realize light weight structures with high damping capacity. In this paper, two damping treatments were examined to improve the damping capacity of steel structures: one is to fill the structure with resin concrete⁽²⁾ and the other is to overlay the structure with fiber reinforced plastics(FRP). The damping characteristics of the composite structure beams under the bending vibration were predicted using the logarithmic decrements of the steel pipes and the high damping materials on the basis of the strain energy of the each material. Moreover, composite structure beams were made to examine the damping characteristics experimentally. The analytical results and experimental ones were compared and were found to be in fairly good agreement. Observations were made on the effectiveness of the high damping materials in improving the damping capacity of structures.

Analysis of Damping Capacity

Figure 1 shows a composite structure beam in which a steel pipe is filled with resin concrete and is also overlaid with FRP on its outer surface. Though the two damping treatments were examined separately in this paper, the model of three layers is used here to generalize the problem. In the analysis, it is assumed that slides between the materials do not occur and that the cross section of the beam remains flat when the beam is under the bending vibration as shown in Fig.2. The maximum change in the potential energy of the i th material from its potential energy at the static-equilibrium position is expressed as follows:

$$U_i = \frac{E_i I_i}{2 \left(\sum_{j=1}^3 E_j I_j \right)^2} \int_0^l M_x^2 dx \quad (1)$$

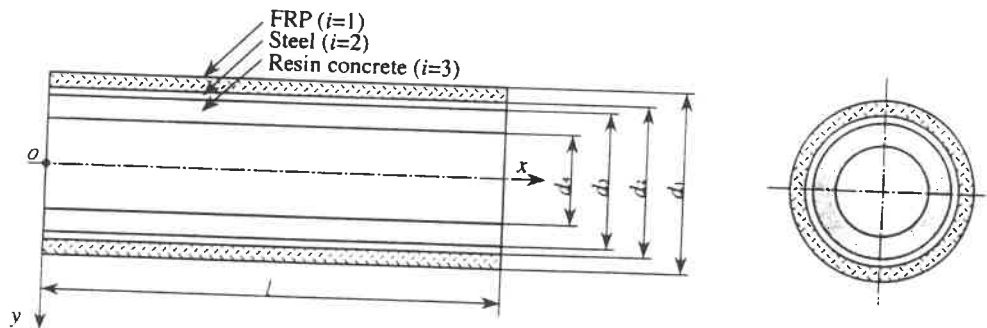


Fig.1 Composite structure beam

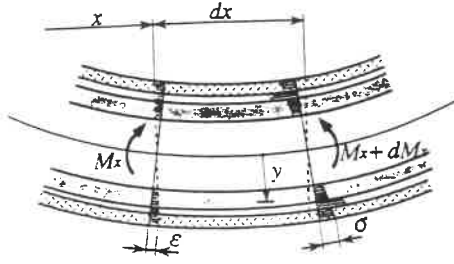


Fig.2 Composite structure beam under bending vibration

Table 1 Properties of materials

Material	Properties	Young's modulus GPa	Density kg/m ³	Logarithmic decrement
Steel		170.9	7.56E+3	8.86E-4
Resin concrete		22.7	2.19E+3	4.40E-2
FRP		16.1	1.60E+3	2.98E-2

where E_i and I_i are the modulus of elasticity and the area moment of inertia respectively. M_x is the bending moment at the position x . i is the material number designating that 1=FRP, 2=steel and 3=resin concrete. The energy dissipated per cycle in the i th material is

$$D_i = 2U_i\delta_i \quad (2)$$

The logarithmic decrement δ of the composite structure beam is written as $\delta = D / 2U$ using the energy dissipated per cycle D and the maximum change in the potential energy U . Thus, from Eqs. (1) and (2) δ is written as follows:

$$\delta = \frac{D}{2U} = \frac{\sum_{i=1}^3 D_i}{2 \sum_{i=1}^3 U_i} = \frac{\sum_{i=1}^3 U_i \delta_i}{\sum_{i=1}^3 U_i} = \frac{\sum_{i=1}^3 E_i I_i \delta_i}{\sum_{i=1}^3 E_i I_i} \quad (3)$$

Finally, the ratio of the logarithmic decrement of the composite structure to that of the original steel structure is obtained as follows:

$$\frac{\delta}{\delta_2} = \frac{1 + (\alpha^4 - 1)\kappa_1\lambda_1 + \{(1 - \gamma^4)\kappa_2\lambda_2 - 1\}\beta^4}{1 + (\alpha^4 - 1)\kappa_1 + \{(1 - \gamma^4)\kappa_2 - 1\}\beta^4} \quad (4)$$

where

$$\alpha = d_1 / d_2, \beta = d_3 / d_2, \gamma = d_4 / d_3, \kappa_1 = E_1 / E_2, \kappa_2 = E_3 / E_2, \lambda_1 = \delta_1 / \delta_2, \lambda_2 = \delta_3 / \delta_2$$

The equivalent bending stiffness $(EI)_{eq}$ and the equivalent mass per unit length $(\rho A)_{eq}$ of the composite structure beam are

$$(EI)_{eq} = \frac{\pi}{64} [1 + (\alpha^4 - 1)\kappa_1 + \{(1 - \gamma^4)\kappa_2 - 1\}\beta^4] d_2^4 E_2 \quad (5)$$

$$(\rho A)_{eq} = \frac{\pi}{4} [1 + (\alpha^2 - 1)\mu_1 + \{(1 - \gamma^2)\mu_2 - 1\}\beta^2] d_2^2 \rho_2 \quad (6)$$

where

$$\mu_1 = \rho_1 / \rho_2, \mu_2 = \rho_3 / \rho_2 \quad (\rho_i \text{ is the density of the } i \text{th material})$$

The natural frequency ν of the first mode of the bending vibration is obtained by the following equation.

$$\nu = \frac{4.73^2}{2\pi\ell^2} \sqrt{\frac{(EI)_{eq}}{(\rho A)_{eq}}} \quad (7)$$

Experimental Procedure

Dimensions of steel pipes used in experiments are 400mm in length and 50.8mm in outer diameter. Wall thicknesses were varied at 2.3, 4.2 and 10.6mm, in which cases the values β are 0.909, 0.835 and 0.583 respectively. Two damping treatments to improve the damping capacity

of the steel pipes were tested. Firstly, the steel pipes were filled with resin concrete. Secondly, pipes overlaid with FRP were made by overlaying with roving cloth immersed in epoxy resin. The properties of the materials used in the experiment are summarized in Table 1.

The composite structure beam was suspended at the nodal points of the first mode of the bending vibration by thin wires and the first mode of the bending vibration was excited by hitting the beam with an impulse hammer. The free vibration was detected by an accelerometer and was stored in the memory of an FFT analyzer. The data were transferred to a PC and the logarithmic decrement was calculated.

Comparison of Analytical Results with Experimental Ones

Figure 3 shows results regarding the logarithmic decrement δ and the natural frequency ν of composite structures. δ and ν are normalized by the logarithmic decrement δ_2 and the natural frequency ν_0 of steel pipes respectively. Steel pipes are fully filled with resin concrete in the case of filling and are overlaid with FRP so that $\alpha = 1.2$ in the case of overlaying. With the increase of the value β , which means the wall thickness of a steel pipe is getting thin, δ becomes larger and approaches to the logarithmic decrement of the high damping materials. This is because the strain energy of the high damping material becomes higher with the decrease of the wall thickness of the steel pipe. Therefore, more energy is dissipated in the high damping materials. It should be noted that the percentage of the strain energy of high damping materials in composite structures must be high enough in order to effectively improve the damping capacity by combining with high damping materials. The analytical results show fairly good agreement with experimental ones.

As shown in Fig.3 (b), the natural frequency ν of the composite structure is largely decreased compared with the natural frequency ν_0 of the steel pipe when the wall thickness of the steel pipe is thin. The reason for this is that the increase of the weight due to the damping treatments is marked compared with the increase of the bending stiffness. If an existing steel structure needs to be modified, overlaying it with FRP is preferable from the view point of lightness of structures.

Light Weight Oriented Design

The damping treatments for existing steel structures studied above increase the weight of structures. In order to realize light weight structures with high damping capacity, it is better to redesign structures considering the bending stiffness and the weight of original steel structures. Though the design method is not shown here, examples of composite structure beams that have the same bending stiffness as an original steel structure are shown in Fig.4. Example I is the

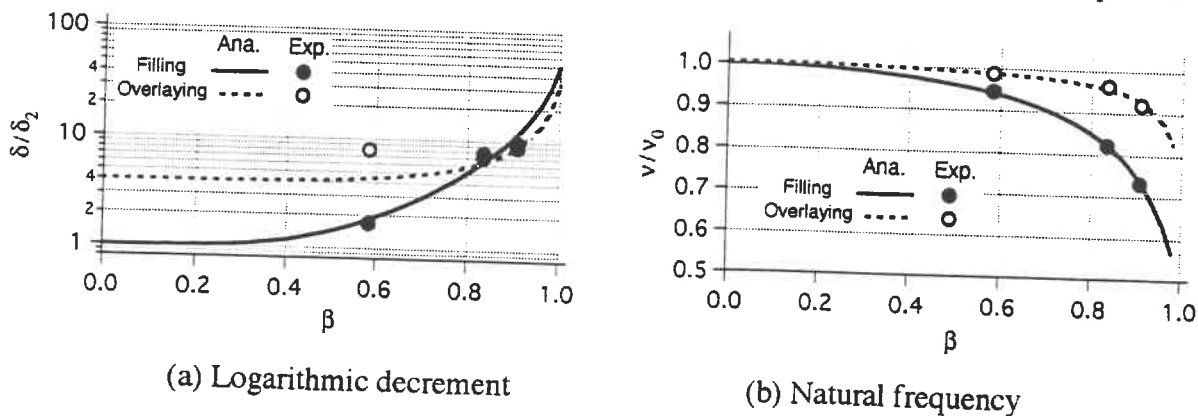


Fig.3 Comparisons of analytical results with experimental ones
(Filling: $\alpha = 1.0$, $\gamma = 0$, Overlaying: $\alpha = 1.2$, $\gamma = 1.0$)

original steel structure of which $\beta_0=0.835$. Example II is a steel-resin concrete composite structure in which the center of resin concrete is hollow to reduce weight. Example III is a FRP-steel composite structure that has the same weight as Example I. In Example II and III, the outer diameter is increased by 30% compared with Example I to obtain the same bending stiffness as Example I. Furthermore, Example IV is a FRP-steel composite structure of which weight is reduced to 75% of Example I. The weight and the damping capacity of these examples compared with those of the original structures are summarized in Table 2. Example II shows a high damping capacity compared with Example III. This is mainly due to the thin thickness of steel and the higher damping capacity of resin concrete than FRP. In the case of overlaying, if the outer diameter is permitted to increase further, light weight structures with higher damping capacity can be realized like Example IV.

Table 2 Characteristics of composite structures

Normalized characteristics	II	III	IV
$(\rho A)_{eq}/(\rho A)_{eq0}$	1.0	1.0	0.75
δ/δ_0	24.7	10.9	15.9

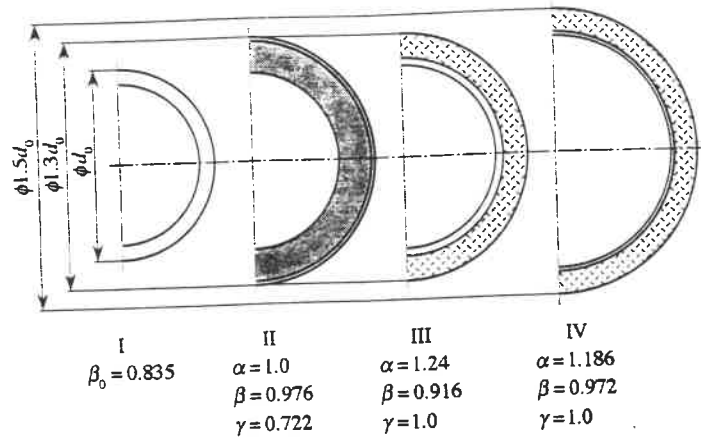


Fig.4 Light weight oriented design examples of composite structures

Conclusions

In this paper, damping treatments to improve the damping capacity of steel structures by combining with high damping materials are discussed. Obtained results are summarized as follows:

- (1) A calculation method of the damping capacity of composite structures was proposed. Results obtained by the method were in fairly good agreement with experimental ones.
- (2) The damping capacity of steel structures is markedly improved by combining with high damping materials only when the wall thicknesses of steel structures are thin. High damping materials should have high modulus of elasticity along with high damping capacity. If an existing steel structure needs to be modified, overlaying it with FRP is preferable as the decrease of the natural frequency is less.
- (3) High damping structures which have almost the same weight as steel structures can be realized with a reasonable increase in dimension.

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Control of a Rotating Head Micro-Tilt Stage for Ultra-Precision Single-Point Diamond Turning

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1. Introduction

In order to perform ultra-precision diamond turning of optical surfaces, higher accuracy and stiffness of workpiece spindle are desirable, particularly in respect of machining "brittle" materials with sub-micrometre cuts [1]. For correcting the orientation of parts in-process whilst machining, a controlled chuck/face-plate micro-tilt stage has been developed. Three optical fibre sensors monitor three degrees of freedom (axial motion and tilts about two axes orthogonal to this). This paper addresses mechanisms, measurements, a control system required to make automatic adjustments to a test-piece on a facing lathe spindle, and the results of performance tests on the system.

2. Mechanical Arrangement

2.1 General

Fig.1 shows the general view of the mechanical arrangement designed for experiments with a controlled tilt-stage. The tilt-stage is mounted on a Type 4R aerostatic bearing (made by Professional Instruments, Minneapolis, MN), and driven by a DC motor via a V-belt on pulleys. The angular position of the spindle required for tilt-stage control is provided by a resolver and voltage signals to piezo actuators driving the stage are provided through mercury slip-rings. The position of the workpiece in respect of three degrees of freedom is monitored by three optical fibre sensors (measuring axially).

2.2 Micro-tilt stage

Fig.2 shows the cross-sectional view of the stage. A previously developed piezo-driven tilt-stage [2] was supported by three leaf springs.

However, from the requirements for higher stiffness and resonant frequency, in the current arrangement, a stage supported by six leaf springs was designed. Further effort was made to minimize static and dynamic cross-coupling between the three degrees of freedom. The characteristics of the developed tilt stage are :

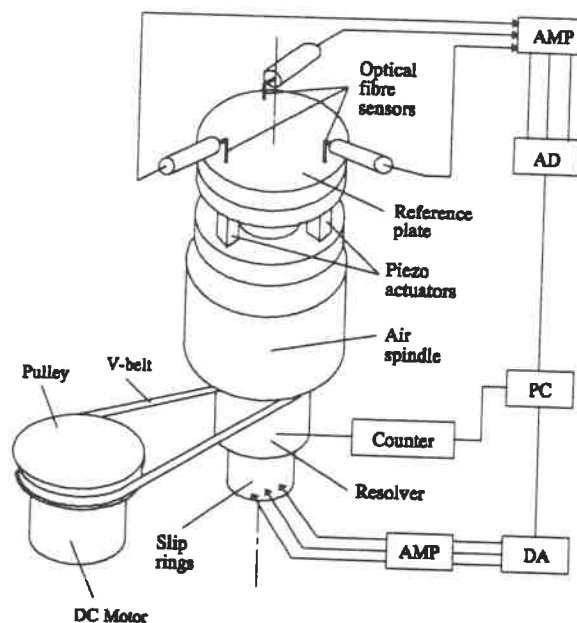


Fig.1 General view of the micro-tilt stage system

Travel range : 8 μm (Axial) 40 arcsec (Tilt)
 Resolution : 10 nm (Axial) 0.05 arcsec (Tilt)
 Stiffness : 196 N/ μm (Axial)
 77 Nmm/ μm (Tilt)
 Resonant frequency : 3 kHz (Axial) 3 kHz (Tilt)

2.3 Optical Fibre Sensors

Optical interferometry has traditionally been used when the test-piece is stationary, and also the implications of dynamic interferometric monitoring of a rotating test-piece have been considered [2,3]. However, due to the requirement of faster measurement, optical fibre sensors (made by Opto Acoustic Sensors, Raleigh, NC) having faster response were used in this work.

3. Control System

Fig.3 shows the dual coordinate systems employed, i.e. fixed coordinates XYZ based on the machine frame containing the three optical fibre sensors and rotating coordinates X'Y'Z based on the tilt stage. Fig.4 shows the control system block diagram. Control inputs are required position values at the three optical fibre sensors, which normally need to be constant, and the control outputs are the three measured positions. Control is required to keep outputs equal to inputs. The controller elements include a repetitive controller, coordinate transfer matrix, PID controller and low pass filter. The control plants, i.e. the rotating tilt-stage is shown by the

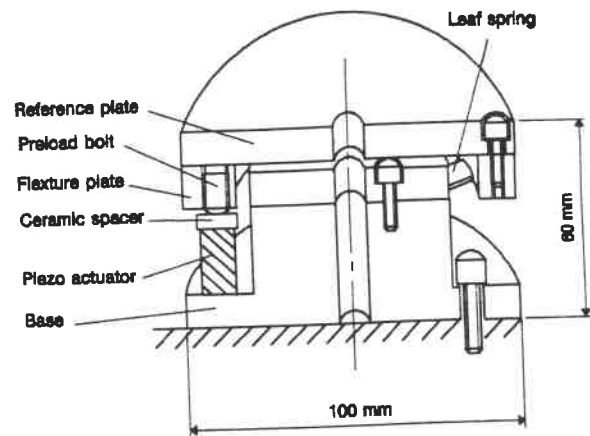


Fig.2 Cross section view of the tilt stage

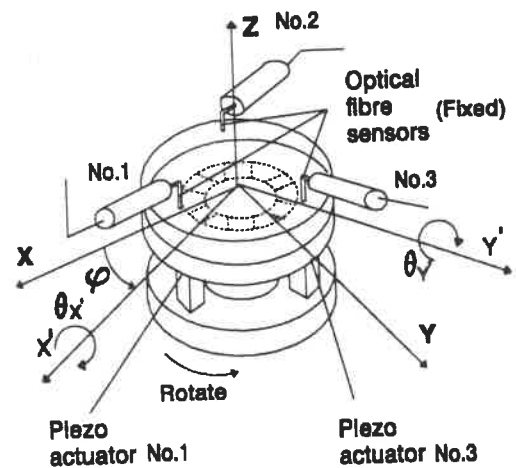


Fig.3 Coordinate system

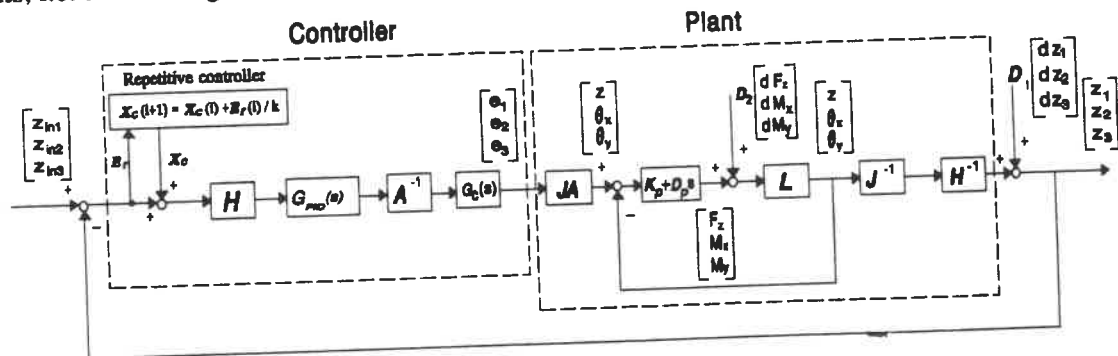


Fig.4 Control system block diagram

coordinate transfer matrix J , piezo constant matrix A , piezo stiffness matrix K_p , piezo damping matrix D_p , and L which shows the characteristics of the leaf spring structure, etc. Two types of disturbance need to be considered. One is the intrinsic rotating error of the spindle, D_1 in the figure, which would be basically repeatable; the other is the error mainly caused by external disturbances, e.g, cutting force, D_2 in the figure.

4. Control experiments

Fig.5 shows the stepwise positioning in (a) the axial direction and (b) the tilt direction with the micro-tilt stage stationary. The results indicate that the controlled micro-tilt stage is capable of positioning over $7\text{ }\mu\text{m}$ with 10 nm resolution in the axial direction and over 35 arcsec with 0.05 arcsec resolution in tilt.

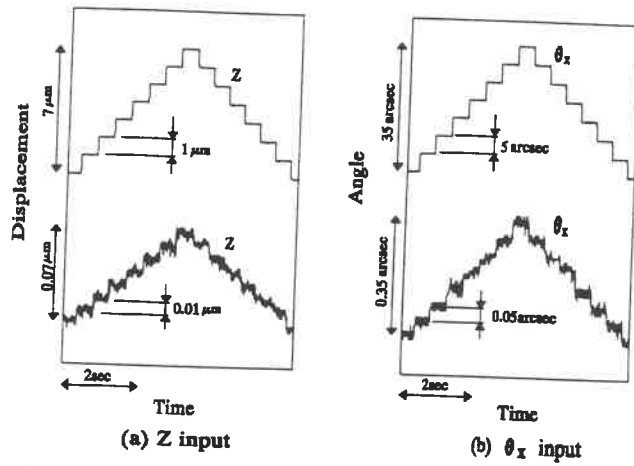


Fig.5 Stepwise positioning with the micro-tilt stage

Fig.6 shows tracking performance of the stationary micro-tilt stage with a PID and repetitive controller, for $4\text{ }\mu\text{m}$ peak-to-peak and 20 arcsec peak-to-peak sinusoidal input signals in each degree of freedom, where motions of the micro-tilt stage and tracking errors from the target motions are shown. Each degree of freedom is controlled to within $\pm 20\text{ nm}$ or $\pm 0.1\text{ arcsec}$ of tracking errors and cross-coupling.

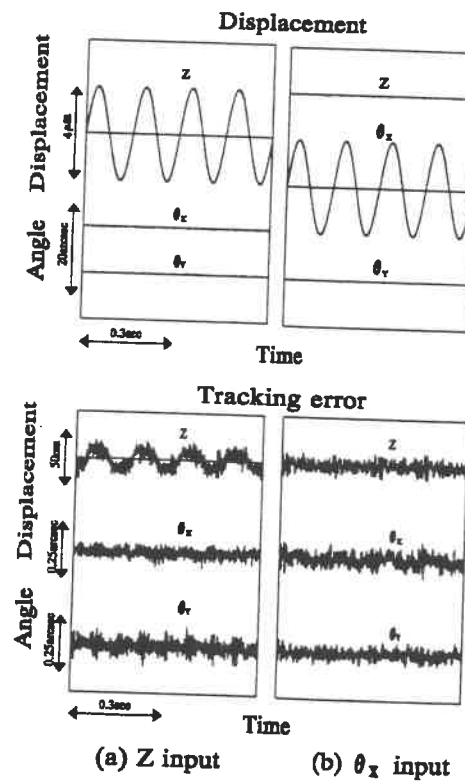


Fig.6 Tracking performance with the micro-tilt stage

Fig.7 shows (a) motion accuracies of the rotating spindle measured at the surface of the reference plate with the optical fibre sensors at a rotational speed of 600 rpm, where averaging over 100 revolutions and spacial filtering are employed in order to reduce random noise caused by the influence of the surface texture of the reference plate in the displacement measurement. Although the controlled displacements are averaged values, where real time control is not undertaken, (b) the spindle motion controlled by a PID and repetitive controller indicates that the micro-tilt stage is capable of reducing motion error during rotation by a factor of about three.

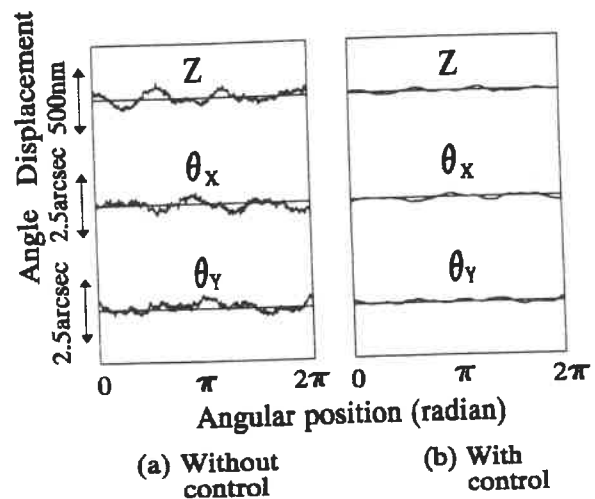


Fig.7 Motion accuracy of the rotating spindle (600 rpm)

5. Summary

A micro-tilt stage and its control system have been designed and developed. The performance of the system has been tested and its application to diamond turning will be examined and developed further.

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Vibration Isolation of Precision Equipment

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Introduction. Continuous tightening of machining and measuring tolerances for parts of machines and instruments leads to development of ever more accurate machine tools and measurement apparatus. Magnitudes of the tolerances are expressed in fractions of micrometers or in nanometers. It was widely accepted from the time of World War II that precision machinery and instruments must be isolated from external vibrations whose amplitudes may exceed significantly the magnitudes of allowable machining/measurement deviations. To satisfy this obvious need, a wide inventory of passive vibration isolation mounts was developed and is commercially available (e.g., [1], [2]). Recently, this inventory was complemented by actively controlled isolators which can provide even better vibration protection and/or maintaining constant level of the device mounted on soft isolating mounts regardless of changes of mass distribution within the machine [2], [3].

Improvements in the vibration isolation efficiency can be achieved by using softer isolating mounts, but the soft mounting is causing static and dynamic instability. Static instability is a change of levelling when mass distribution within the isolated system is changing due to addition/subtraction of components to the system or due to movements of massive parts in the system. Dynamic instability is rocking of the isolated system due to both external excitations (touching of the isolated body, air drafts, etc.) and internal excitations (force pulses due to acceleration/deceleration of the moving components). These rocking motions are undesirable since they may induce displacements in the working zone of the system (e.g., cutting area or measurement area), possibly even exceeding the external vibration-caused displacements from which the system is being isolated. Both static and dynamic instabilities can be alleviated by the appropriate active (servo-controlled) systems, but such systems add significantly to the cost and dimensions of the isolated object. In addition, its reliability can be reduced due to addition of complex electronic and/or pneumatic control systems.

The intent of this paper is to demonstrate that in many cases vibration isolation specifications for high precision objects can be not as stringent as it is usually assumed since sensitivity to external vibrations is not necessarily increasing with the increasing accuracy requirements, and since quality of vibration isolation can be in many cases decoupled from the issues of static and/or dynamic stability by a proper understanding of mechanisms of effects of the external vibrations on the object performance and also by a proper understanding of dynamics of the vibration isolation system.

Ambient Vibrations. Vibration isolators are usually intended to reduce the floor-transmitted vibrations. Floor vibrations are excited by vibratory motions of nearby processing machines, motors, transformers, etc. (e.g., as the result of fast rotating unbalanced rotors, chatter vibrations in metal cutting machine tools), by passing-by carts, dollies, cranes, etc.; by shocks from impact-acting machines, such as forging hammers and presses, from machines with reversible masses, such as surface grinders, from accidental impacts during material handling, from footsteps on non-rigid floors. For ultra-precision facilities, microseismic motions of the ground could be also significant excitations. The character and levels of floor vibrations depend not only on the excitation, but also on the filtering effects of the dynamic system consisting of the soil, foundations, floor, and other building structural components.

Floor vibrations usually are made up of a multitude of spectral components with randomly varying amplitudes. Fig. 1 shows a summary of an extensive study of quasi-steady floor vibrations at the various manufacturing plants in the former Soviet Union in the 1960s. The data was collected from time domain vibration plots (oscillogramms); maximum amplitudes of vibratory processes which were reasonably stable for 5-10 cycles were recorded. The broken lines are envelopes of points representing absolute maxima at various frequencies; the solid lines are assumed boundaries of the spectra which were used in analytical derivations. These lines are representative for vibrations at the ground floors (solid foundation slabs). On upper floors made of thin prestressed concrete slabs, the amplitudes tend to be 1.5 to 3.0 times higher and the corresponding frequency ranges tend to be broader.

Dots on the plots in Fig. 1 represent recent measurements at the U.S. manufacturing plants. The data was obtained using FFT analyzers, presumably using windows prescribed for steady-state processes, thus the amplitudes for the lower end of the frequency spectrum seem to be exaggerated. These low-frequency amplitudes are comparable with amplitudes of transient vibrations of the floor, Fig. 2.

However, the most interesting mid-range of the spectrum (10-30 Hz) shows that the old Russian data gives a conservative upper limit of floor vibrations at the U.S. manufacturing sites equipped with more powerful and faster modern machinery. The main difference is in the upper frequency limit of relatively intense vibrations, which is much lower for the recent tests at the U.S. sites (~20 Hz) than for the old Russian tests (30-40 Hz). The floor vibration surveys are usually concerned only with vertical vibrations, while majority of precision equipment are more sensitive to horizontal vibrations. Maximum amplitudes of horizontal vibrations are 25-30% lower than

maximum amplitudes of vertical vibrations and their frequency range is about 30% narrower (upper frequency of high amplitudes frequency range is 20-30 Hz vs. 30-40 Hz for vertical vibrations).

Intensity of floor vibrations in precision microelectronics manufacturing facilities is much lower. Vibration spectra of vertical vibrations for three typical clean room floors in [11] indicate maximum levels of vibratory velocity (60-200 $\mu\text{m/s}$) are in 10-60 Hz range, with the corresponding displacement amplitudes not exceeding 2 μm (0.05 μm).

Detrimental Effects of Floor Vibrations on Precision Equipment. There are several ways in which floor vibrations can disrupt operation of precision equipment. One of the most important is their tendency to destabilize the dimensional set-up chain. Due to inevitable friction, the elements of such chains (e.g., guides, screws, arms, clamps) are fastened in elastically strained conditions. Vibrations lead to oscillations of the joint contact pressure and thus to frictional force variations, leading to relaxation of the elastic strains and to possible "jumps" in the dimensional chain. Such jumps may produce steps in machined surfaces, changes in dimensions of automatically machined parts, or misalignments in optical systems. Vibration-caused "jumps" at joints of mounting systems (e.g., wedges, jack-screws) may lead to deformations in the machine bed, which in turn may result in reduced accuracy, chatter, and increased wear.

Floor vibrations are also inducing relative motions between the cutting or measuring tool and the workpiece. Such motions may cause waviness of the machined surface, out-of-roundness, or a deterioration in surface finish (especially for turning machines, where the waviness during successive revolutions is phase-shifted). Impulsive vibrations can lead to local variations in the depth of cut which result in dents on the machined surfaces.

Vibrations in the measuring zone, between the probe or the light source and the measured surface, are blurring the measurement results and are effectively reducing resolution of the measuring instrument.

Model of Vibration Transmission. Floor vibrations are detrimental for performance of precision equipment since they excite relative vibrations in the working (cutting)/measurement zone or in the joints comprising the dimensional set-up chain. Effects of floor vibrations on machine tools usually are significant at the lowest, seldom also at the second, natural frequency of the dynamic system, largely because the higher natural frequencies lie considerably above the frequency range of significant excitation amplitudes. The first natural frequency for grinders is typically between 30 and 70 Hz, the second - between 100 and 150 Hz.

Fig. 3 from [4] is a typical vibration sensitivity plot for precision production equipment for microelectronics. Minima represent structural natural frequencies of the devices (marked on the figures). It can be seen that the lowest natural frequencies are between 4 and 20 Hz, and many of the higher natural frequencies lie between 30 and 70 Hz, i.e., in the frequency range of the reduced floor amplitudes (see Fig. 1).

Dynamics of vibration transmission from the floor into the working zone of a precision equipment unit can be approximately represented by three two-mass systems like the one in Fig. 4, one for each coordinate direction of the floor vibrations. Masses and springs in the model are representing generalized inertias and stiffnesses of the unit, their values usually different for each coordinate direction. M_b represents mass of the bed, M_u - mass of its upper structure (tool head or measuring head), k_m, c_m - equivalent stiffness and damping of the structural components and joints, and k_v, c_v - stiffness and damping of the mounting (e.g., jacks or isolators). The effect of vibrations on an equipment unit represented by such a two-mass model may be investigated in terms of the ratio of the relative displacement amplitude X_{rel} between masses M_u and M_b to the displacement amplitude X_b of the bed M_b .

Because M_b, M_u and k_m are generalized parameters, X_{rel} also is a generalized relative displacement. The actual relative displacement in the working area of the equipment unit is proportional to X_{rel} , and the proportionality constant γ (design constant) depends on geometry of the machine. For example, for an upper structure that can be represented by model in Fig. 5,

$$\begin{aligned}\gamma_{zx} &= mcd/I_{O1} = cd/(r_y^2 + b^2 + c^2) \\ \gamma_{zz} &= mcb_1/I_{O1} = cb_1/(r_y^2 + b^2 + c^2)\end{aligned}\quad (1)$$

where the first subscript on γ refers to direction of the bed vibration and the second to direction of motion of the upper structure relative to the bed. Here m denotes the actual mass of the upper structure, I_{O1} its moment of inertia about its actual axis of rotation, and r_y radius of gyration about the principal y-axis. The dimensions b, b_1, c and d , as indicated in Fig. 5, are measured from the elastic center O_1 of the system. (Note: the elastic center is the point about which the elastic restoring forces produce zero net moment if the system is moved without rotation.). The machine set up and position of the upper structure with respect to the bed influence γ . The case in Fig. 5 is a good approximation for structural designs of numerous machine tools and measuring systems. Vibration sensitivity of such model can be reduced by reducing structural (design) dimensions c, b_1, d . Experiments on numerous machine tools of similar types have shown that their vibration sensitivity may vary as much as 30 times depending on design.

Principles and Criteria of Vibration Isolation. Principles and criteria of vibration isolation of high-precision equipment can be considered for three typical cases. The first case deals with protection of a specific unit of equipment from specific steady-state vibrations of the floor. Since the steady-state vibrations can be

represented by a discrete spectrum, this case is, essentially, a case of isolation of the unit of equipment from harmonic vibrations of the floor. *The second case* represents protection of a specific unit of equipment from any combination of typical quasi-steady state floor vibrations within the envelope in Fig. 1. Although the plots in Fig. 1 present a frequency range of vibrations, at a particular site only some spectral components may have amplitudes as high as the plot indicates. If a vibration isolation system protecting the unit of equipment from any realization of the floor vibration spectra represented by plots in Fig. 1a, b is to be developed, then the unit can be equipped with isolating mounts which would assure its specified performance for the majority of installation sites. *The third case* deals with protection of equipment from transient motions of the floor caused by regular or accidental shock excitations.

For the most important *second case* the principles involved in isolation of a precision machine against floor vibrations are illustrated in Fig. 6 [5]. The bottom-most graph here shows the spectrum of floor displacement amplitudes X_f in a given direction. This graph is a streamlined envelope of the plots in Fig. 1 for the considered direction. The top-most graph shows the corresponding relative motion in the machine's work area, with Δ_0 representing the limit of acceptability of such motions. Dynamic coupling between the machine structure and the vibration isolation system in precision machines can be assumed to be relatively weak. Consequently, the approximate relative motion graph can be obtained simply by multiplying together the X_{rel}/X_b , X_b/X_f and X_f graphs, and the vibration isolation criterion can be expressed as

$$a_0 \frac{X_b}{X_f} \frac{X_{rel}}{X_b} < \Delta_0, \text{ or } a_0 \frac{X_b}{X_f} \mu(f) < \Delta_0 \quad (2)$$

The solid lines in Fig. 6 correspond to a stiff mounting system under the machine, the dashed lines to a softer isolation system with the same damping as the stiff system, and the dotted lines to a softer isolation system with an increased damping. The set of graphs in the left half of Fig. 6 pertains to viscous damping in the isolators, for which greater damping results in greater vibration transmission at frequencies above $\sqrt{2}f_v$; the graphs on the right side pertain to hysteretic damping, which affects the response at high frequencies only insignificantly. It is clear that relative vibrations in the work area are reduced by use of a softer mounting with greater damping. *Resonance on vibration isolators is an allowable regime*, and the isolators must be selected in such a manner that amplitude of relative vibrations X_{rel} at the resonance of the vibration isolation system does not exceed the specified level Δ_0 . Accordingly, improvements in isolation can be achieved by reducing f_v and/or by increasing damping (δ_v).

Use of generalized spectrum of floor vibrations, as in Fig. 6, is a more conservative approach than using actual vibration measurements at the installation site. The latter can change after relocation of old and installation of new equipment, while the spectrum assumed in Fig. 6 represents the worst case based on multiple measurements as given in Fig. 1.

Performance of an isolation system can be characterized by the "isolation criterion"

$$\Phi = \frac{f_v}{\sqrt{\delta_v}} \leq \sqrt{\Delta_0 f^2 / \pi a_0(f) \mu(f)} \quad (3)$$

where δ_v denotes log decrement of the isolation system, $a_0(f)$ is amplitude of the floor vibrations at various frequencies (the bottom-most graph in Fig. 6). Different values of Φ apply for different coordinate directions. If vibration sensitivity $\mu(f)$ of a precision object is known (for example from the plots in Fig. 3), then expression (3) can be used for specifying vibration isolation parameters. For the case of Fig. 3, $\Delta_0 = 0.1 \mu m$ is specified. Table 1 lists required values of Φ for the sensitivity peaks from Fig. 3.

The minimum required Φ for vertical direction is 4.51 Hz which corresponds to $f_v = 3.04$ Hz for medium-damped isolators, $\delta_v = 0.6$, and $f_v = 5.0$ Hz for $\delta_v = 1.2$. For horizontal directions $\Phi_{min} = 6.05$, and $f_v = 4.7$ Hz for $\delta_v = 0.6$, $f_v = 6.7$ for $\delta_v = 1.2$. The natural frequencies for high damping are realizable with passive isolators.

Similar analyses for other three high precision instruments have shown that only one actually requires an active vibration isolation system for protection from the typical spectrum of *industrial* steady floor vibrations (minimum Φ less than 0.8 Hz). However, such ultra-precision units are never used in the general industrial buildings. "Vibration criterion" curves [4] require vibration velocity levels of the floors for special precision facilities between 2,000 $\mu in/sec$ (the least demanding criterion VC-A) and 125 μin (the most demanding criterion VC-E) in the frequency range 8-80 Hz. This corresponds to maximum displacement amplitudes (at 8 Hz) between 1 μm for VC-A and 0.06 μm for VC-E. For such environments the required isolation criterion values would be much higher than specified in Table 1, 1.7 times higher if the building complies with VC-A and 7 times higher if the building complies with VC-E. Accordingly, much stiffer passive isolators could be used in most of the considered cases.

The third case of vibration isolation involves isolation from short duration (impulsive or shock) motions of the floor. Such excitations tend to be less troublesome, because of their short durations. Accordingly, one may

take the permissible peak relative displacement Δ_{op} in response to floor shocks as 3 times the value of Δ_0 permissible for steady vibrations. One may analyze an isolation system for a machine subject to shock excitation by use of the modified shock spectrum approach that has been described in the context of forging hammer isolation [6]. In most of cases, required parameters of the vibration isolation systems determined for steady floor vibrations would be also adequate for isolation of impulsive vibrations.

Vibration Isolation System. Required parameters of vibration isolation systems for various units of precision equipment, as presented in Table 1, are calculated with the assumption that the unit is mounted in such a way that there is no dynamic coupling between the vertical and horizontal vibrations of the unit. But, due to substantially nonuniform mass distribution in typical equipment units, difficulties in determining center-of-gravity positions and calculating weight distribution between the mounting points, large differences in nominal stiffnesses between sequential models of vibration isolators in a given design line, and large deviations of the stiffness values from the nominal values due to manufacturing tolerances, installation of precision machines on conventional constant-stiffness isolators tends to result in a strong coupling of vibrations in the various directions.

This problem can be alleviated considerably by use of constant-natural-frequency (CNF) isolators, in which stiffness in both vertical and horizontal directions is proportional to the weight load on the isolator (e.g., [7]). As a result, such isolators provide a high degree of dynamic decoupling without a need to determine the center-of-gravity position, to calculate weight load distribution between the mounting points, etc. In addition to this, such isolators have a significantly reduced sensitivity to manufacturing tolerances [7].

Effect of using CNF isolators is illustrated by Fig. 7, which shows the frequency response of the relative motion between the grinding wheel and the table of a surface grinder due to vertical floor vibrations of 5 μm amplitude. It may be noted that use of CNF isolators providing natural frequency in vertical (z) direction of 20 Hz resulted in much better isolation than even the best constant-stiffness isolators with the substantially lower 15 Hz resonance frequency in the z-direction. The difference between natural frequencies of 15 and 20 Hz is equivalent to approximately two times difference in isolator stiffness.

Production models of CNF isolators for vertical natural frequencies 10, 15, 20, 30 Hz have been developed. Recent research results obtained in the WSU Machine Tool Research Laboratory have provided a basis for designing CNF isolators for very low vertical natural frequencies as low as 1-2 Hz.

Side Issues for Vibration Isolated Equipment. Vibration isolation using passive isolators results in a reduced stiffness of connection between the isolated equipment and the rigid floor (foundation). This can lead to rocking motions of the equipment if it contains internal massive moving units (e.g., tables), and to reduction of effective rigidity of the equipment structure which is not tied to the rigid foundation.

The rocking can be reduced by using massive foundation blocks and placing isolation under these blocks. This approach tends to be expensive, however, and makes relocation of the machine very difficult. However, in many cases satisfactory results can be achieved, alternatively, by using anisotropic isolators and/or by increasing distances between isolators in the direction of the internal excitations.

Smaller precision machines usually do not require addition of a foundation for rigidization, whereas larger machines usually do, unless they are specially designed to be mounted on three points (kinematic mounting). In many cases, judicious placement of the isolation mounts may reduce static deflections of the machine bed, thus in effect making it to act as if it were more rigid. This was demonstrated on many practical installations. Because the proper selection of the number and the location of the mounting points is important both for reducing the rocking and for increasing the effective rigidity of the machine structure, this problem generally deserves particular attention. Designers rarely give this the consideration it deserves.

Conclusions

1. Parameters of floor vibrations in various manufacturing plants fit into "constant displacement amplitude" patterns for a limited frequency range (different for vertical and horizontal vibrations). Use of these patterns allow to formulate objective (usually, conservative) specifications for parameters of vibration isolation systems.
2. Sensitivity of production and measuring equipment to external vibrations can be significantly reduced by a thoughtful structural design.
3. Natural frequencies of vibration isolation systems for precision equipment units (and, consequently, stiffnesses of vibration isolators) can be increased by enhancing damping of isolators.
4. Many high-precision production and measuring machines can be successfully installed on passive vibration isolators having high damping, with active pneumatic/electronic isolators being needed only in exceptional cases.
5. Constant Natural Frequency (CNF) isolators can substantially simplify design of a vibration-isolated installation and, in the same time, noticeably improve its performance.
6. Rocking and reduction of effective stiffness of vibration-isolated machines can be significantly alleviated by judicious selection of number and locations of isolators as well as of their stiffness characteristics.

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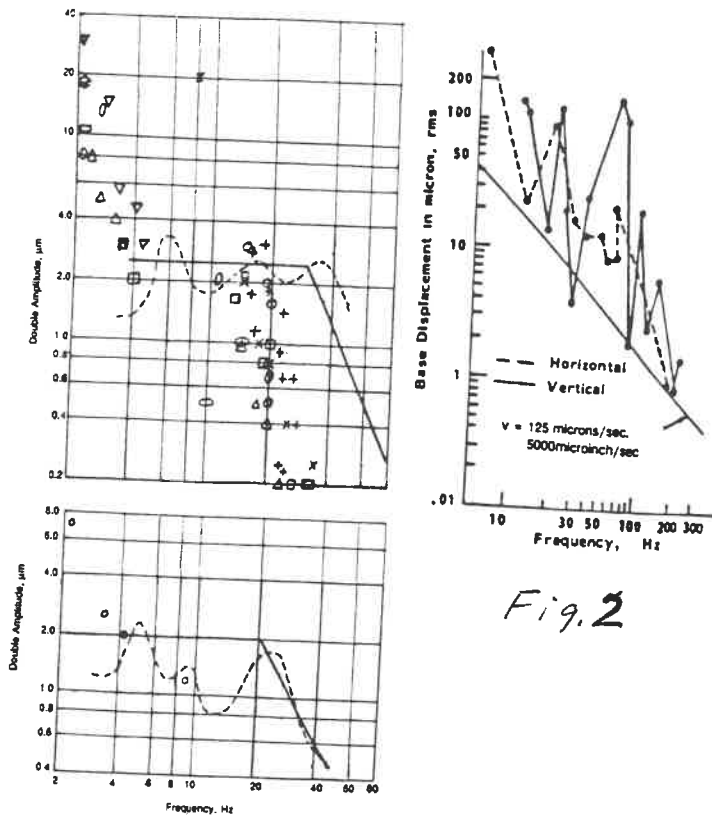


Fig. 2

Fig. 1

Table 1. Vibration Isolation Criterion for case of Fig. 3

a. Vertical direction

f, Hz	11	12	20	25	30	32	41	70	80
$\mu(f)$	0.0083	0.010	0.087	0.0091	0.056	0.303	0.05	0.0077	0.010
Φ , Hz	4.51	12.3	7.0	26.9	13.0	6.3	22.5	128	137

f, Hz	95	100	110	140	150	160	220	250
$\mu(f)$	0.588	0.294	0.037	0.455	0.222	0.2	1.28	0.77
Φ , Hz	23.1	35.3	92.5	47.0	74.7	86.7	55.2	86.3

b. Horizontal direction

f, Hz	7	12	22	65	70	100	140	210
$\mu(f)$	0.0033	0.05	0.0125	0.071	0.090	0.090	0.056	1.25
Φ , Hz	13.7	6.05	22.3	49.6	49.2	84.0	176	68.6

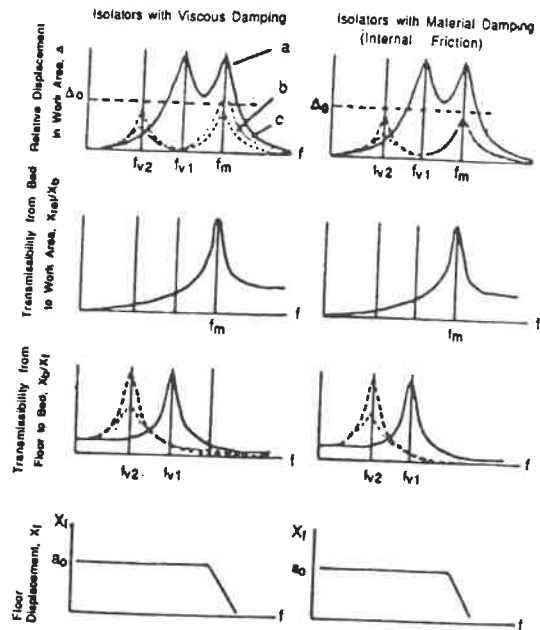


Fig. 5

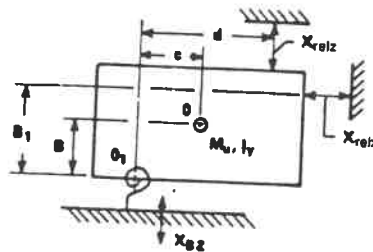


Fig. 4

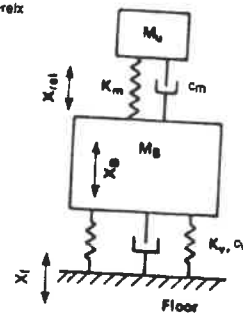


Fig. 3

Design of Linear Piezomotor Positioning Apparatus for Precision Manufacturing Applications

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1. Introduction

Linear piezomotors have been used for precision positioning applications because they have high positioning resolution, high stiffness, unlimited travel range with no backlash and stick-slip. Progress on the development of the linear piezomotors has been achieved for the last decade with most efforts concentrated in Japan[1]. Various designs have been disclosed on the linear piezomotors, however, most of them have small output power and very low travel speeds. There are four types of linear piezomotors[2] as shown in Fig.1. Burleigh Instruments, Inc. has designed an inchworm type piezomotor which can carry a load of several pounds with a nanoprecision resolution[3]. The first two types of piezomotors have been successfully applied to STM. Researchers at Tokyo Institute of Technology developed two types of piezomotors which can travel at the maximum speed of 50 cm/s[4], but the load capacity of such types of piezomotors are low. Due to their small output power and low travel speed, the existing piezomotors have very limited applications. For most of the precision manufacturing applications, a linear piezomotor with large output power and high travel speed is to be developed.

2. Design of the Linear Piezomotor

A new linear piezomotor was designed for a precision positioning apparatus. This new design allows the piezomotor to have large output power, high travel speed, high precision (with submicron resolution), unlimited travel distance, and high stiffness. The new design is shown in Fig.2. It is constructed by a frame accommodating three piezoelectric actuators and a rail. Two of the actuators act as clamps and one as extension element. The frame is so designed that it can eliminate any possible

shearing and/or tensile forces from the piezoelectric actuators, and integrate all the actuators as a whole.

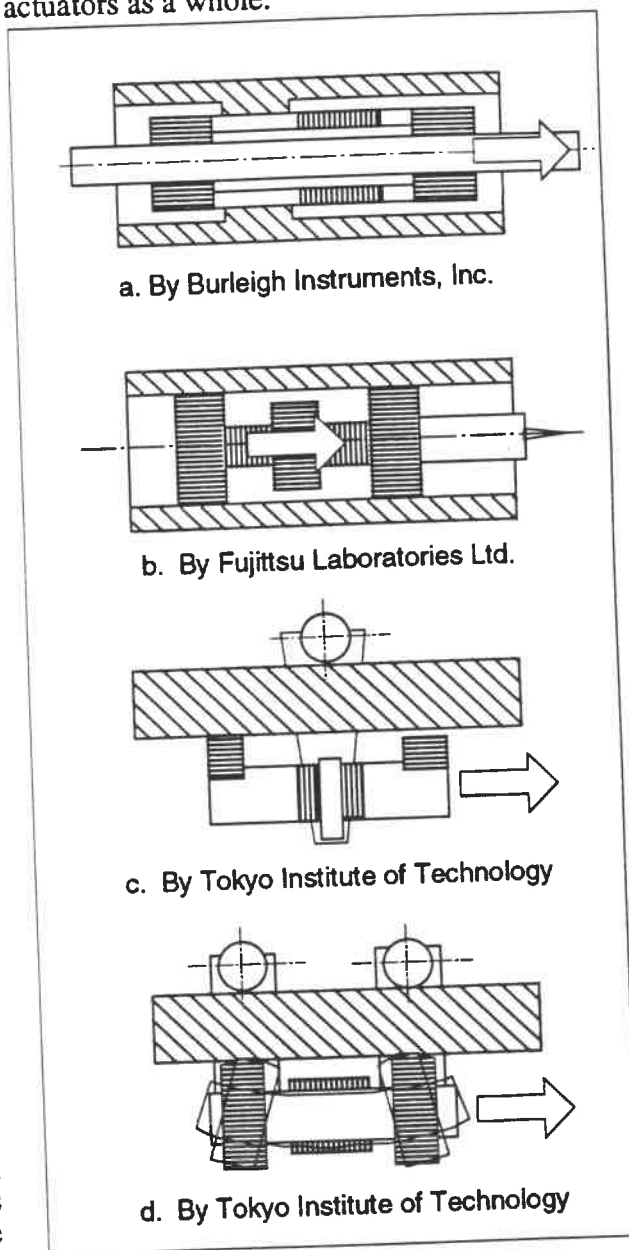


Fig.1 Linear piezomotors developed during recent years

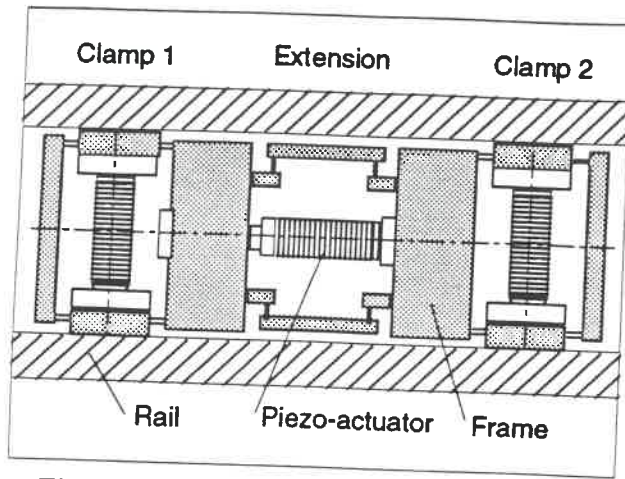


Fig. 2 Newly designed linear piezomotor

a. Motion Analysis

In dealing the motion analysis, the linear piezomotor can be simplified as a multi-degree-of-freedom vibration system shown in Fig.3. Frictions at the interface of clamps and rails, and damping at the interface of piezoactuator-frame and clamp-rail should be considered in terms of dashpot. This system can be treated separately in X- and Y-directions. The extension movement in X-direction is related to two clamping movements through the clamping and frictional forces between clamps and rails. The equation of motion in X-direction is:

$$[m_x]\{\ddot{x}(t)\} + [c_x]\{\dot{x}(t)\} + [k_x]\{x(t)\} = \{Q_x(t)\}$$

where $[m_x]$ is a diagonal inertia matrix, $\{Q_x(t)\}$ the force vector with piezoactuating forces and clamp-rail frictional forces as its components. The stiffness matrix and damping matrix are symmetric and follow as:

$$[k_x] = \begin{bmatrix} 2k_1 + 2k_2 & -2k_2 & 0 & 0 & 0 & 0 \\ -2k_2 & 2k_2 + k_3 + k_5 & -k_3 & 0 & -k_5 & 0 \\ 0 & -k_3 & k_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & k_4 & -k_4 & 0 \\ 0 & -k_5 & 0 & -k_4 & k_4 + k_5 + 2k_6 & -2k_6 \\ 0 & 0 & 0 & 0 & -2k_6 & 2k_6 \end{bmatrix}$$

$$[c_x] = \begin{bmatrix} 2c_1 & 0 & 0 & 0 & 0 & 0 \\ 0 & c_3 & -c_3 & 0 & 0 & 0 \\ 0 & -c_3 & c_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & c_4 & -c_4 & 0 \\ 0 & 0 & 0 & -c_4 & c_4 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$$

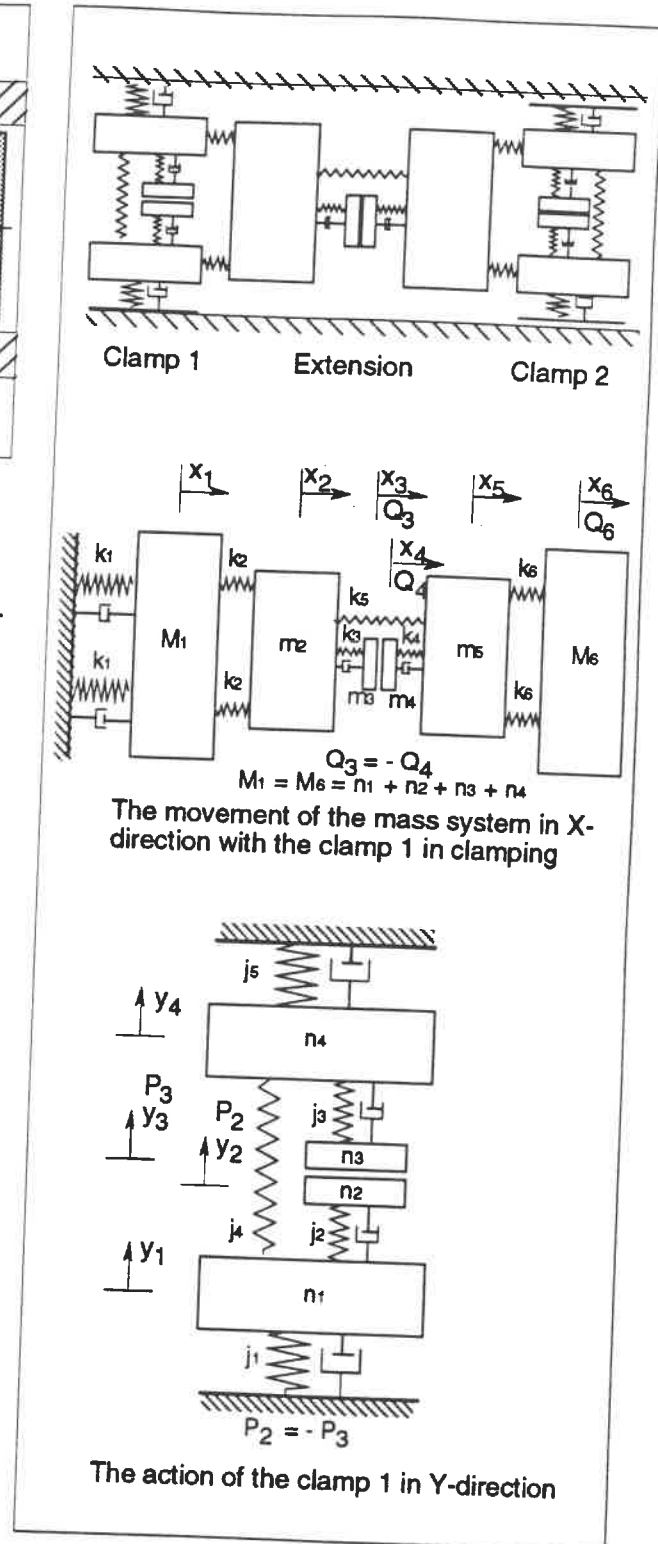


Fig.3 A multi-degree-of-freedom system

The clamping movement in Y-direction is described by the following equation:

$$[m_y]\{\ddot{y}(t)\} + [c_y]\{\dot{y}(t)\} + [k_y]\{y(t)\} = \{Q_y(t)\}$$

where,

$$[m_y] = \begin{bmatrix} j_1 + j_2 + j_4 & -j_2 & 0 & -j_4 \\ -j_2 & j_2 & 0 & 0 \\ 0 & 0 & j_3 & -j_3 \\ -j_4 & 0 & -j_3 & j_3 + j_4 + j_5 \end{bmatrix}$$

$$[c_y] = \begin{bmatrix} c_1 + c_2 & -c_2 & 0 & 0 \\ -c_2 & c_2 & 0 & 0 \\ 0 & 0 & c_3 & -c_3 \\ 0 & 0 & -c_3 & c_3 + c_5 \end{bmatrix}$$

Since there are two clamps in Y-direction which have different actuating forces and motions, two sets of equations are needed to describe the motions of the two clamps. Finally, the motion of the linear piezomotor is given by the following three sets, totally 14 equations:

$$[m_x]\{\ddot{x}(t)\} + [c_x]\{\dot{x}(t)\} + [k_x]\{x(t)\} = \{Q_x(t)\}$$

$$[m_{y_p}]\{\ddot{y}_p(t)\} + [c_{y_p}]\{\dot{y}_p(t)\} + [k_{y_p}]\{y_p(t)\} = \{Q_{y_p}(t)\}$$

$$[m_{y_q}]\{\ddot{y}_q(t)\} + [c_{y_q}]\{\dot{y}_q(t)\} + [k_{y_q}]\{y_q(t)\} = \{Q_{y_q}(t)\}$$

Where $\{y_p(t)\}$, $\{y_q(t)\}$, and $\{x(t)\}$ are coordinates of the first clamp, second clamp in Y-direction and the whole system in X-direction.

b. Stress-Strain Analysis

The stress-strain analysis in frame design was accomplished by FEM. The flexure hinges for clamping and unclamping actions provide a stiffness of 20 N/μm in the direction of clamping motion, and 1,000 N/μm in the direction of travel motion, while the flexure hinges for extension action in the direction of travel motion provide a stiffness of 37 N/μm which is lower than that of the piezoelectric actuator (55 N/μm). The maximum pushing force of these actuators is 1,000 N. The actuators with a larger pushing force of up to 30,000 N and a higher stiffness of up to 700 N/μm may be used for special applications where high stiffness and precision are needed. These actuators are preloaded by a kinematic support so that frame may not be over constrained and distorted. The distance

between the two internal walls of the rail is critical and is adjustable in this design. The gap between the wall and the clamp shoe is in the range of 0 - 5 μm. The tolerance between two walls of the rail is smaller than that of the maximum clamping travel. A successful linear piezomotor with a maximum clamping motion of 3 μm has been reported by the STM researchers.

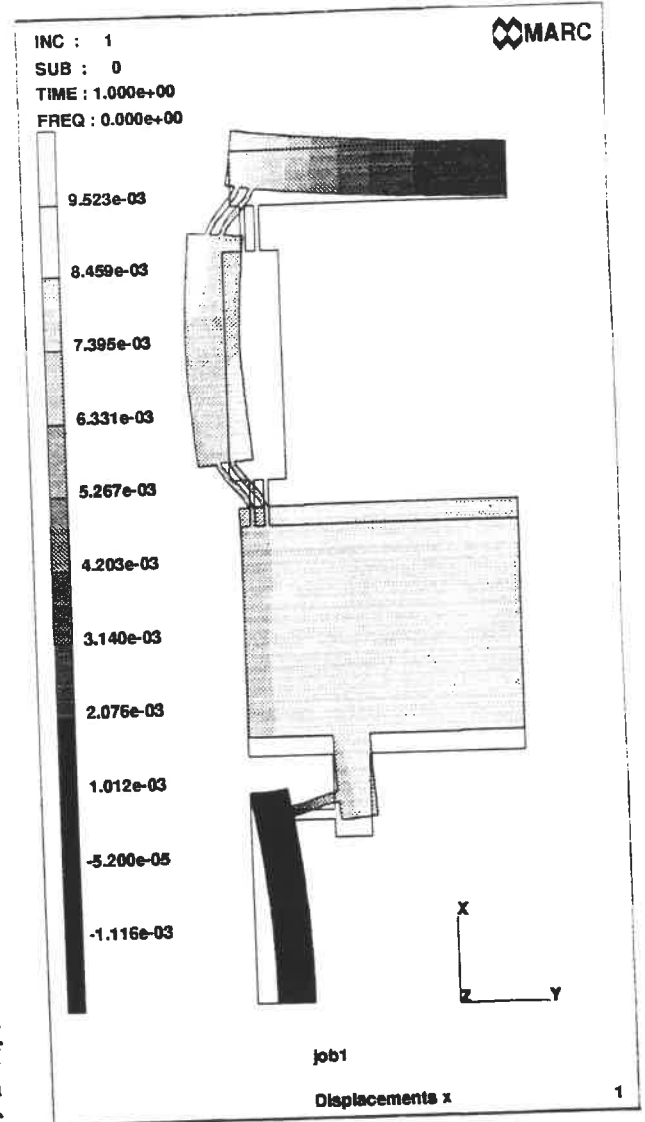


Fig.4 Elastic deformation of the frame under 800 N clamping and 600 N extension forces (showing a quarter of the frame)

3. Properties of the Positioning Apparatus

In the present design, the rail has a large stiffness and precise wall surfaces. Efforts have been made towards lessening the

maximum clamping motion though the piezoelectric actuators used for the clamps allow us to have a maximum expansion of 15 μm . The maximum extension of the clamping motion is 8 μm when the actuator is given an extension of 15 μm . The slow-motion constant-velocity property may be insured by the high stiffness, and the large output of this design. Piezoelectric actuators have an advantage of maintaining their positions without consuming energy. This will result in a reduced heat generation when a piezomotor is at rest. Heat generation occurs when the motor is in operation. The friction at the interfaces of the rail and the clamp should be as large as possible so as to achieve large output power. Unlike that in conventional driving systems such as a ball screw system, the friction does not give negative effects against motion. The wear occurring on either the rails or the clamps after a long time of operation, will be compensated by increasing the clamping stroke, so that the performance of this positioning apparatus will not be affected. If nanoprecision positioning is targeted, materials with a low coefficient of thermal expansion should be used. Magnetic fields will not affect the performance of the linear piezomotor since the whole unit can be fabricated with non-magnetic materials. This is another unique property of the piezoelectric motors. Fabrication of this linear piezomotor positioning apparatus can be cost-effective since the apparatus contains less elements. For workshop environment applications, a high resolution grating interferometer with a ceramic scale can be used.

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DYNAMIC COMPENSATION OF MEASURING MACHINES

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The use of computers to compensate for repeatable geometric errors in machines was first developed by NIST in the 1970's. These concepts were refined and implemented in production CMMs in the mid 80's. The impact on the industry was immediate. Today nearly every measuring machine being produced uses microprocessor enhanced accuracy techniques. The resulting improvements to industrial metrology have extended beyond better accuracy and include stable performance for extended periods. The next most significant challenge facing the industrial metrology industry is to be able to provide the same level of accuracy and stability in shop environments as measuring machines are capable of providing in temperature controlled laboratories.

Many machine shops have relatively poor environmental controls and temperature variations of 30°C are not uncommon. It is widely known that changes in temperature affect measurement accuracy but the cost and inconvenience of putting the measuring equipment in a temperature controlled room limits the practice. This creates a problem for the manufacturing engineer who needs to make good measurements for accurate process control.

Current State-of-the-Art

Efforts being made to overcome errors caused by temperature variations rely on passive methods and involve three basic approaches. The first and most common method is to control the flow of heat into the machine. This is most often done by providing temperature controlled rooms. Other techniques involve controlling warm-up cycles, the use of air and oil showers and providing insulation as appropriate. The second approach is to incorporate compensation techniques. The compensation algorithms are either developed by mathematics using FEA (Finite Element Analysis) or through empirical testing. The infinite number of variable temperatures that occur within the CMM when the temperature is changing limits the effectiveness of compensation in factory environments. For every machine there will be some construction differences. It is difficult to estimate precisely the boundary conditions at each assembly joint. The third approach involves the designing of measuring machines to minimize their sensitivity to

temperature changes. Thin walled aluminum has been widely used because of its high rate of thermal conductivity. Increasing efforts have been made to use the same material throughout the machine to avoid bi-metal effects. In some cases coolant is circulated throughout the machine to keep its temperature constant while the temperature of its environment is changing. Most recently there is a growing interest in using materials having very low temperature coefficients of expansion. Examples of these are carbon fiber, ceramics, Invar and Zerodur.

While these approaches are implemented to reduce the effects of temperature they do not fundamentally compensate for the machines' distortions as the temperature is changing throughout its structure. There is a need for maintaining machine accuracy independent of its mechanical frame.

Statement of Problem

The metrology frame of a CMM distorts as a function of temperature changes and gradients. This is true whether the metrology frame of the CMM is self contained or part of a foundation for a large gantry style machine. These infinitely variable errors created by changes in machine geometry could be avoided if microprocessor enhanced accuracy concepts were applied in real time. This is the objective of an Advanced Technology Program(ATP) contract awarded to Giddings & Lewis earlier this year. The ATP contract is being administered by NIST and is supported by research from the University of Michigan

Recommended Approach

To accomplish the objectives of the contract the metrology frame of the measuring machine will be made up of laser beams rather than to use the bearing way surfaces as has been used on every machine ever produced. Sensors will be used to detect minute variations between the bearing way surface and the laser beam. These spatial variations will be sensed in their independent degrees of freedom consisting of three translational and three rotational motions. Just as parameters are inserted to the equations of motion for today's microprocessor enhanced accuracy techniques as a means of calibrating the measuring machine in the manufacturer's plant, this same process will be performed in real time while the machine is being used on the factory floor.

Since the laser beam has been selected as the reference, a recent effort has been made to improve its pointing stability. In non-interferometric laser metrology, the laser beam is usually used as the measurement reference. However, it is well known that the measurement system based on this type of laser reference can only achieve limited

accuracy due to the beam drift of typical He-Ne laser sources. In recent research done at the University of Michigan, it has been demonstrated that single mode optical fibers can be used as a unique beam conditioning device. After coupled into a single mode optical fiber, the beam exhibits a surprisingly high stability. Both lateral and angular drifts of a typical He-Ne laser beam have been greatly reduced. Additional advantages of such an optical fiber beam delivery system include its low cost and the nature of being a cold light source that generates almost no heat.

The first practical application of this concept will be on large gantry machines along the major axis where the machine is most sensitive to geometric changes from stress and temperature changes. Results of the implementation of real time accuracy enhancement on gantry machines are most encouraging and provide the basis for positive projections of improvements when applied to all three axes of a typical CMM.

Development Plan

The accuracy of a CMM is defined by the positioning accuracy of a probe in a 3-dimensional working volume. The volumetric accuracy is influenced by the geometric errors of each of the three moving axes of the CMM and by the non-orthogonality between any pair of the three axes. Most current state-of-the-art CMMs are equipped with some types of geometric error compensation algorithms. Giddings & Lewis is a world leader in the development and implementation of geometric error compensation techniques for CMM applications. It has implemented a so-called MEA (Microprocessor Enhanced Accuracy) technology on all its CMM product lines. [Bell et al., 1989 and 1990] The fundamental assumption implied in these geometric error compensation techniques is that the geometric error information, gathered ahead of time before a CMM is shipped to its customer, will remain unchanged during the field use. This is true when the CMM is to be used in a well protected environment with temperature control and in the case of a large gantry style machine, when the foundation is stable. However, in view of the increasing demand for moving CMMs out of the well protected environments and using them directly on the manufacturing shop floor, this fundamental assumption is being challenged. The geometric error information determined at the time of the machine's manufacture will not remain constant when the machine is used in unstable environmental conditions. This will adversely effect the machine's accuracy.

The scope of the research is to develop a sensory system to monitor the change of some dominant geometric error components on-line and will develop a scheme to update the information base in the MEA compensation system so that not only the spatial variant geometric error components but also the temporal variant components are included in the compensation.

The research is being carried out in two phases. In Phase One, the work is focused on the basic sensory system development and on the analytical simulation study of the behaviors of CMM structural elements under unstable environmental conditions. In Phase Two, the emphasis is on the development of an adaptive compensatory system and on the actual implementation and experimental validation of the system.

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Key Words: coordinate measuring, temperature compensation, laser, microprocessor

OPEN ARCHITECTURE CNC: THE PATS CONTROL SYSTEM

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INTRODUCTION

This paper presents a description of the control system for the Precision Automated Turning System (PATS). The open architecture of this system is determined by the requirements placed on the machine and the configuration and characteristics of the final design of the machine. Before the architecture of the controller and a typical application are discussed, it is helpful to present some of the requirements and characteristics of the PATS.

MACHINE REQUIREMENTS

The PATS, shown in Figure 1., is being built as part of the Department of Energy's Precision Flexible Manufacturing System Program. The PATS, with its two machining stations and two inspection stations, was designed to machine and measure components made from hazardous materials. To protect the operators, it was designed to be operated in a complete protective enclosure. These requirements created a demand for numerous special functions to be performed by the control system, in addition to many conventional computer numerical control (CNC) functions. Some of these special functions include:

- The control of ten axes of motion, including the spindle axes
- The simultaneous running of two independent part programs on either of the two machining stations, the two inspection stations, or a machining station and an inspection station
- The simultaneous running of cooperative programs on the two machining stations, (e. g., the part transfer operation)
- The modification of operating part programs based on real time data from the machine and its environment, (e. g., machine temperature compensation, or tool compensation)
- The operation of the different work stations from independent operator stations
- The collection of machining swarf during the machining process.

These functions, in various ways, exceed the capability of most commercially available CNC controllers, and only with great difficulty could they be added to the rigid architecture of these controllers. Another

complicating factor was that the PATS was conceived as a prototype machine to be used in the development of techniques for operator interaction, workpiece handling, waste management, work enclosures, tool management, maintenance and cleaning of the machine, in addition to the development of the actual machining and measuring procedures. Many of these required functions can only be defined once the machine is in operation. The control system has to be flexible enough to accommodate adding and changing functions at a later time.

MACHINE CHARACTERISTICS

The PATS, as shown in Figure 1., has overall dimensions: height = 7.0 feet, width = 14.7 feet, depth = 16.5 feet and weight = approximately 40,000 pounds. The ranges of motion on the linear machining axes are: X & X'--14 inches, and Z & Z'--33 inches. On the inspection axes the ranges are U & U'--10 inches, and W & W'--18 inches. The linear positioning accuracy of all linear axes is ± 40 microinches with a repeatability of ± 10 microinches. The resolution on the position measurement for the linear axes is 4.0 microinches, based on the precision, linear, glass scales used for position feedback. The position control systems have velocity and acceleration feed forward features that when active result in a zero position loop following error if the command signal is smoothly profiled. The maximum feedrate on all linear axes is 30 inches/minute.

The spindles (C & C' axes) can be operated in either the closed loop velocity or position control modes. In the velocity mode, the speed range is 0 to 1500 RPM with a regulation of 1%. In the position mode the resolution is 0.01 degrees and the maximum speed is 16,200 degrees/minute (450 RPM).

The spindle tables (B & B' axes) and the tool changer carousels (not shown on Figure 1.) are servo controlled through the programmed logic controller (PLC) interface. The spindle tables have a position resolution of 0.01 degrees (36 arcseconds) and a maximum rate of rotation of 1440 degrees/minute (4.0 RPM). These axes are held in one of four positions by mechanical detents that maintain their position accuracy to 0.5 arcseconds.

The PATS was designed to machine hemispherical shells whose major diameter is up to 14 inches and whose final thickness is between 0.03 - 0.08 inches. The inner and outer contours of the part must be machined with an accuracy of ± 0.0005 inches.

THE CONTROLLER

The controller has to provide the full range of CNC functions and the special functions listed above. It must also have the flexibility to have its hardware and software systems modified by the users as operating procedures become defined. In the opinion of the authors these requirements dictated an "open architecture" control system. "Open architecture" implies that the controller is based on well defined and documented hardware and software systems that allow the user to modify the control functions. This means the controller should have a commonly accepted bus structure, such as the VME or AT buses, that allows hardware expansion. This also means its computer should run under a commonly available multitasking operating system, such as OS/2 or Windows NT, and its programs should be written in a modular fashion using a higher level language, such as C. Finally, the entire system must be documented so that a user can successfully modify it. Obviously, for a system that does not require the flexibility of the PATS this openness would probably not be necessary and, indeed, if future versions of the PATS were to be built the control system could be some fixed subset of the system we have acquired.

The controller selected was a UNIDEX 31 (made by Aerotech, Inc., of Pittsburgh, PA). A block diagram of the controller and the functional units of the PATS are shown in Figure 2. Some of the functional units are not planned for implementation in the initial configuration, however, provisions had to be made for their potential inclusion.

The bus structure of the UNIDEX 31 is shown in Figure 3. The main part of the controller is packaged in the VME hardware standard. The main computer uses an Intel 486 which runs under the OS/2 operating system. OS/2 is a multitasking operating system which provides a graphical user interface. Disk storage consists of a 200 MB hard drive and a 3.5 inch floppy drive. The main computer communicates with a PC/AT bus which allows network communications to higher level computer systems and provides interfaces to several different PLCs. The PLCs can also be addressed through the VME bus.

The software system is shown graphically in Figure 4. The application code is modularly constructed and written in the C language compiled with the Microsoft C 6.0 compiler. The multitasking feature of the OS/2 operating system allows multiple independent tasks to run simultaneously. Therefore, multiple CNC programs can be assigned to the various machine stations with independent part programs. Interprogram communication is handled by global variables accessible from each part program. OS/2 device drivers are supplied which allow the main 486 CPU to communicate with

the VME and AT buses as well as PLC input/output and the axis processor. All servo control algorithms are fully digital and run on a high speed RISC CPU on the axis processor module. The RISC processor can handle up to 16 coordinated axes with a 4 kHz servo update rate. In addition, the axis processor module can contain up to 8 MB of on-board RAM which is used for real time data acquisition of probe gauging and axis parameters.

THE PART TRANSFER OPERATION

A typical operation requiring the type of controller described above is the automatic part transfer operation. At the completion of the machining of the outer contour, the shell must be transferred to the other spindle for machining of the inner contour. The automatic transfer requires accurate alignment of the two spindles and positioning them so that one chuck can release the shell and the other chuck can receive and hold it in position for the next machining. The spindle alignment accuracy required for this transfer is maintained by the mechanical detents on the B and B' axes. The transfer procedure and required instrumentation are described below.

The transfer process starts when one numerical control part program (NCP) completes the machining and measuring of the outer contour of the shell. At this point a second NCP assumes control of the spindles to directly drive the Z & Z' axes. The drive commands are generated from instrumentation signals measuring the force on the shell either directly through force transducers or indirectly through the drive motor current. As an alternative or supplement to the force measurement, the position of the shell may be measured either from the spindle positions or from a linear variable differential transformer embedded in the fixturing. These procedures have been tested in an experimental part transfer fixture and found to be successful.

This is just one example of the use of the open architecture structure of the controller in the PATS operation. Other examples for the use of this flexibility, such as the integration of linear probes and a vision system for tool tip inspection, will be part of the poster presentation.

CONCLUSIONS

The use of an open architecture controller for the PATS machine provides the flexibility and expandability needed to meet the changing requirements of the overall system.

The use of an open architecture controller should be considered for any machine, or other manufacturing process, if changes to the operation are expected during the useful lifetime of the equipment.

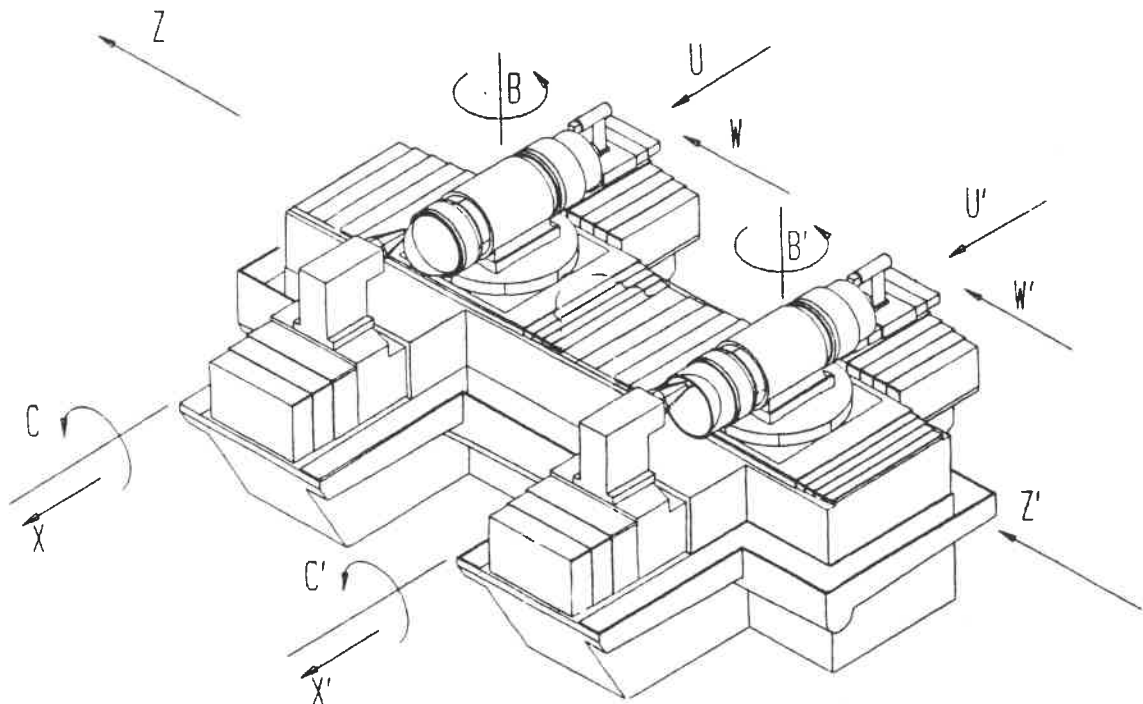


FIGURE 1. THE PRECISION AUTOMATED TURNING SYSTEM

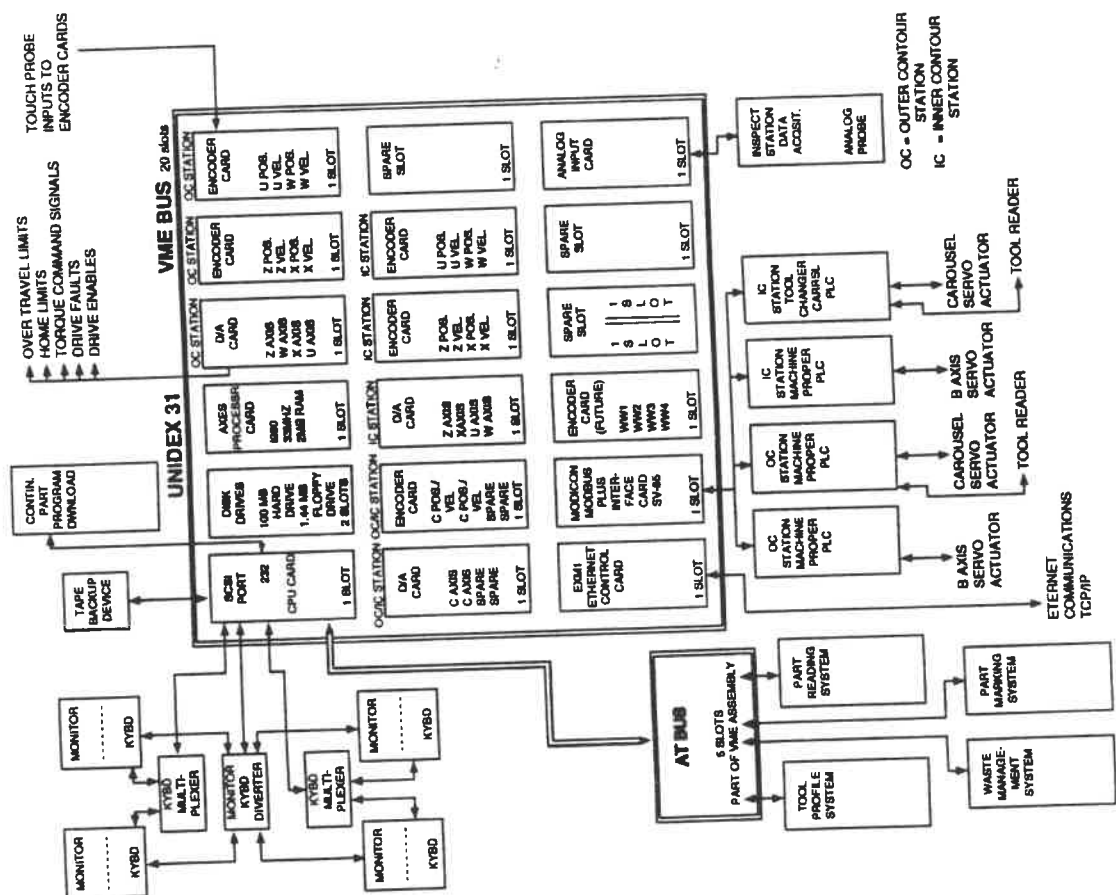


FIGURE 2. PATS CONTROL SYSTEM BLOCK DIAGRAM

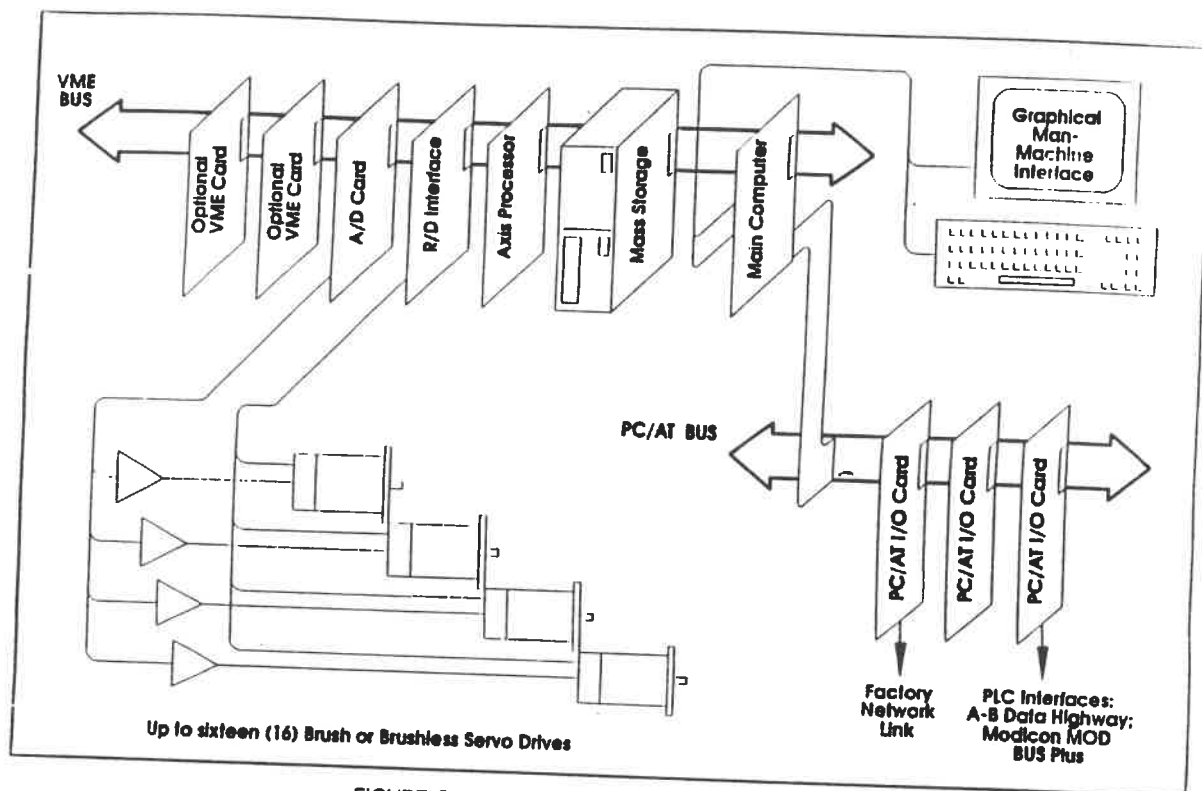


FIGURE 3. UNIDEX 31 BUS STRUCTURE

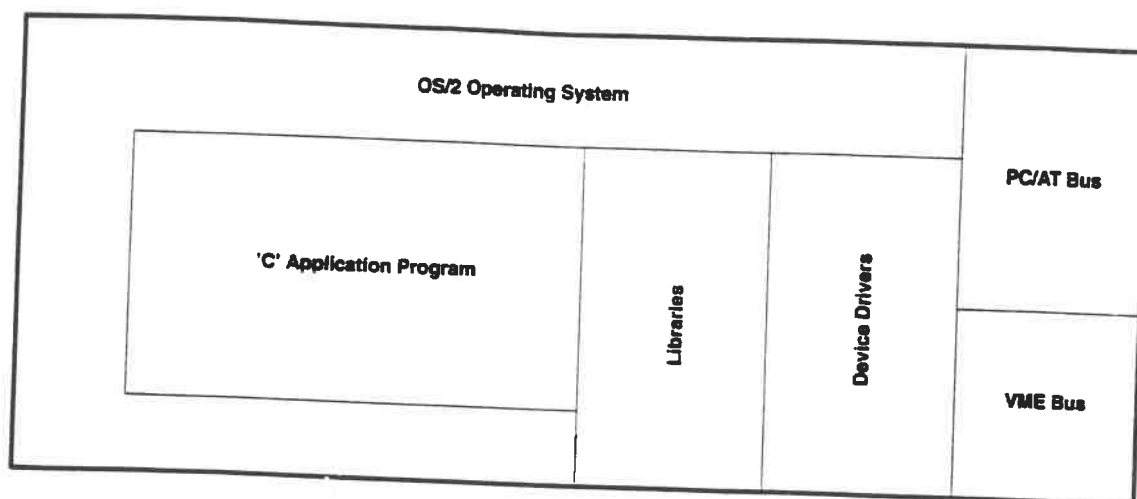


FIGURE 4. UNIDEX 31 SOFTWARE STRUCTURE

ON-LINE DETECTION OF TOOL FAILURE IN TURNING USING ACOUSTIC EMISSION

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ABSTRACT

This paper describes an indirect in-process technique for monitoring the cutting tool condition. An attempt is made here to extract maximum information from Acoustic Emission(AE) signal acquired during machining. Statistical methods like time series modelling technique was used to extract parameters called features representing the state of the cutting process. Autoregressive(AR) model parameters, power of the signal and AR residual signal were investigated and found to show effective and consistent trend towards tool wear. Once all these features were extracted after preliminary processing of the signals, tool status, whether worn or unworn state was decided using a pattern recognition technique.

INTRODUCTION

Sensor based monitoring and diagnostic systems have become increasingly useful in improving the efficiency of manufacturing systems. The efficiency of a manufacturing system is affected by tool failure. Hence, many diagnostic systems have been attempted to monitor tool failure. Further, where optimization schemes based on adaptive control are used, in-depth information, such as estimates of tool wear and wear rate becomes necessary.

A variety of signals such as tool-tip-temperature, cutting forces, torque, power, vibration, Acoustic Emission(AE), etc., have been used for monitoring tool failure. Of these, only a few have been successful, even in limited applications, outside laboratory. Research has turned towards the use of multiple sensors like vibration, cutting forces and AE /1, 2/. The best results seem to be obtainable from tool forces and acoustic emission /3/.

SCOPE OF THE PRESENT WORK

The present work attempts to device schemes for worn tool detection by the measurement of acoustic emission signal. The signals are smoothened for further processing to extract the features. Dynamic signal of AE is subjected to stochastic analysis using time series modeling to generate a residual signal and to calculate Auto Regressive(AR) parameters. Power of the dynamic signal(P_s), power of the residual signal(P_r) and AR parameters are considered as features representing the cutting

parameters. Once these features are computed, decision making regarding tool condition is done through pattern recognition technique.

EXPERIMENTAL DETAILS

Carbon steel (C-60) bars were turned using coated carbide tool. Using a fresh cutting edge, initial experiments were carried out at different cutting conditions. This data was used to determine the limits of machining conditions for further experiments. Experiments were performed to obtain measurements with progressive wear with selected cutting conditions ($v = 300 - 450$ m/min, $s = 0.1$ to 0.16 mm/rev and $a = 1-2$ mm). Depending on the length of cut, machining was stopped after every 60 to 80 seconds and width of the flank wear (V_b) was measured. AE signals were also recorded. The set of measurements taken immediately prior to taking wear measurements were used for training the modelling schemes.

A piezoelectric AE transducer was fixed on the top of the tool holder using couplant. The pencil lead break test was used in the calibration of the system and to estimate the attenuation factor of the AE signal, when the signal is transmitted from the work piece/tool material junction. Electrical signals produced by the transducer were of very low amplitude and high frequency content and were to be first amplified with a low noise pre-amplifier. Most of the pre-amplifiers have a gain of 40-60 db. The pre-amplified signals were collected in the signal analyzer and later transferred to a PC-AT through General Purpose Interface Bus (GPIB) for further processing.

RESULTS AND DISCUSSION

Acoustic emission signal during turning was analyzed in order to study tool wear phenomenon. The sensitivity of AE signal to tool fracture was studied by using quantifiable characters like rise time, RMS voltage, peak amplitude, ring down count, events, energy, etc.

Digitized signal of acoustic emission collected in PC-AT was processed to eliminate unwanted noise and to extract useful information. The signal was smoothened to remove low frequency noise present. Dynamic signal at different stages of wear development can be regarded as a stationary series as it takes only a fraction of second for a set of few hundred data points sampled each time and hence statistical methods for stationary normal processes can be appropriately applied to these dynamic signals. An Auto Regressive model of an appropriate order is fitted to the signal. Selection of AR-order was done using Final Prediction Error (FPE) and Akaike Information Criterion (AIC) /4/. An order of five was chosen for AR modeling. Then the AR-residual series, the difference of the actual observations and the AR-predicted values was generated.

AR model is defined as

$$\hat{x}(n) = a_1x(n-1) + a_2x(n-2) + \dots + a_px(n-p)$$

where, $x(n)$, $n = 1, 2, \dots, N$ is the time series (acoustic emission)
 $\hat{x}(n)$ is the AR - predicted value.
 p is the AR order
 $a_1, a_2, \dots, a_p$ are the AR parameters.

Then the AR residual
 $R(n) = x(n+p) - \hat{x}(n+p)$, $n = 1, 2, \dots, N-p$

Once all the features from the signal were obtained, pattern recognition technique was applied for classification of the tool between two classes, viz., any tool with the flank wear greater than 200 microns was considered as worn state and less than that as sharp state. Feature selection and thereby data reduction to improve computational efficiency were done using class mean scatter criterion.

AR parameters computed from the series, possess considerable information regarding the magnitude as well as frequency distribution of the signal and hence used as features for pattern recognition. Coefficients of AR model of order five a_1, a_2, a_3, a_4, a_5 , were divided by a_5 to obtain A_1, A_2, A_3, A_4 and to reduce feature size.

Figure 1 shows wear curves at different speeds with clearly defined regions of running-in, steady state and rapid wear. Figure 2 shows the power of the dynamic signal which increases with flank wear without uniformity. In Figures 3 & 4, both power of the signal and residual signals are shown. Change in the residual signal reflects the change in the process (tool condition in this case). It represents not only the change in the magnitude but also the change in frequency distribution. Power of the residual signal was found increasing with the flank wear. The variation of power of the residual signal was found very much similar to the variation of power of the dynamic signal, but with minimum fluctuation.

An interesting phenomenon was observed with AR parameters. When normalized AR parameters were plotted, viz., A_1 vs A_2 or A_3 vs A_4 for different cutting conditions (Figures 5 & 6), points were found to be clustered near the origin for sharp edge state. However as tool wears out, the points scatter away. This was due to the increase in number of frequency components in the signal, which was in turn due to the rubbing action caused by excessive wear.

A comparison of performance of classification using pattern recognition technique considering various features are given in Table 1. It can be observed that features of the normalized AR parameters along with cutting conditions gave a percentage of

correct classification of 82.35, where as power of the dynamic signal and residual signal along with cutting conditions gave a percentage of correct classification of 64.75.

CONCLUSIONS

Experimental work was designed to use AE sensor and to obtain sensory data for sharp tool and for different stages of the flank wear. Variations in material properties have been identified as a major noise source in signals measured during machining. pre processing of the AE signals was done to eliminate noise and to extract relevant information. Through time series modeling, AR parameters were extracted and taken as features. Power of the signal and residual signal were found effective and consistent. Tool failure detection using Pattern recognition technique was implemented. Performance of the detecting system based on different features were compared.

Table 1 Pattern recognition analysis results of AE signal

FEATURES	PERCENTAGE OF CORRECT CLASSIFICATION
V, s, a, A ₁ , A ₂ , A ₃ , A ₄	82.35
V, s, a, P _s ,	64.75
V, s, a, P _r	69.53
V, s, a, A ₁ , A ₄ , P _s , P _r	73.52
V, s, a, A ₁ , A ₃ , A ₄ , P _s , P _r	67.64

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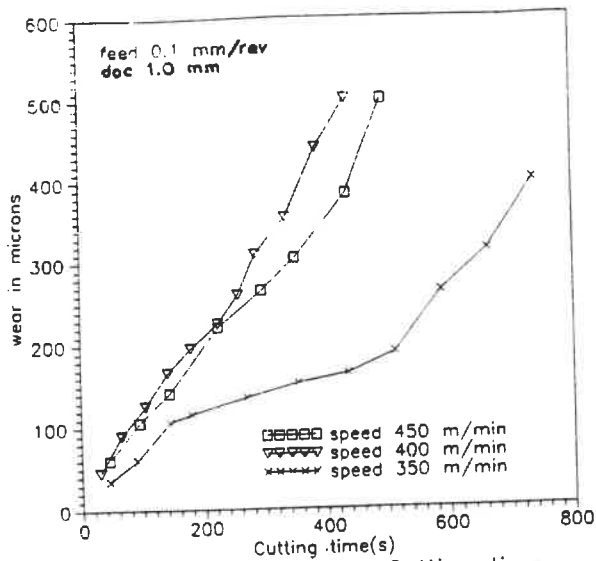


Fig. 1 Flank wear vs Cutting time

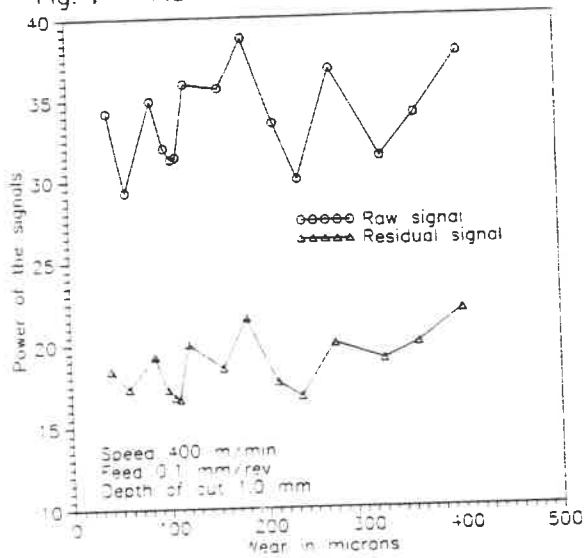


Fig. 3 Power of the signals vs wear

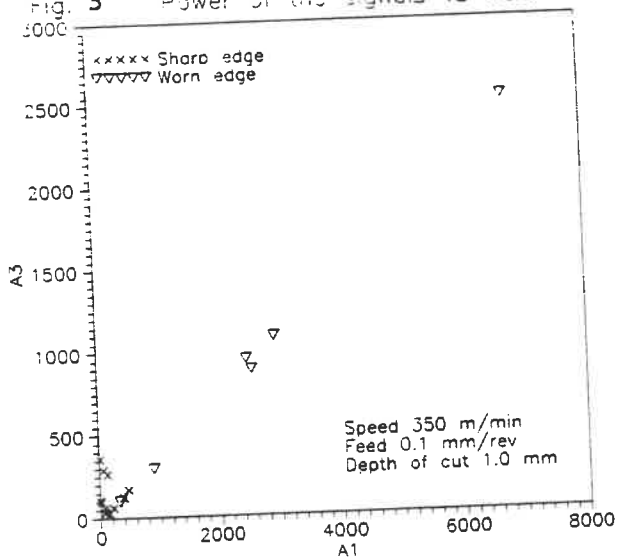


Fig. 5 Modified ratio of AR parameters

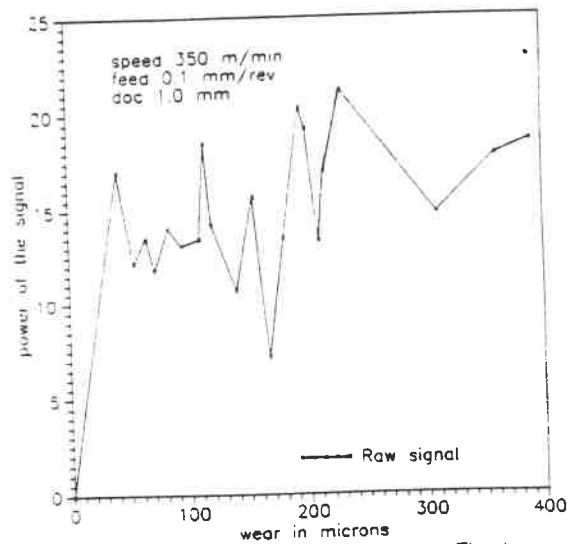


Fig. 2 Power of the raw signal vs Flank wear

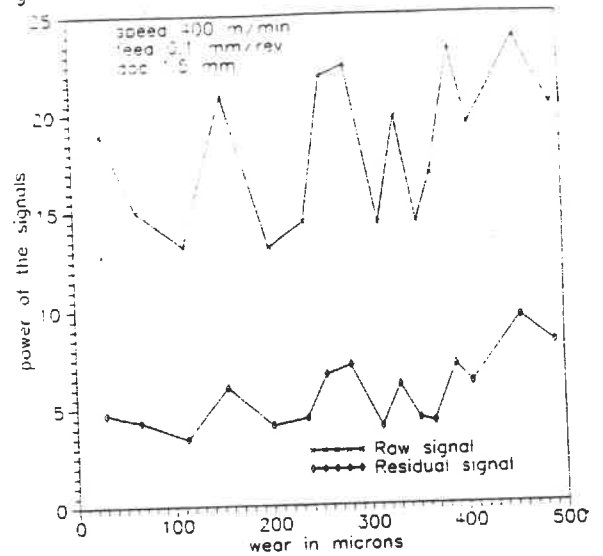


Fig. 4 power of the signals vs wear

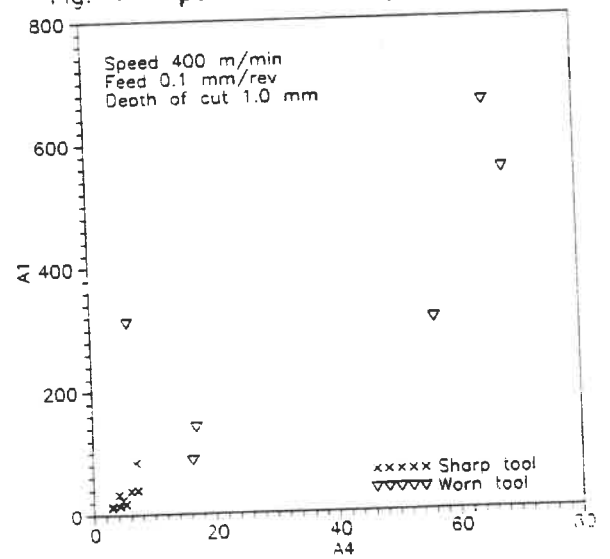


Fig. 6 Modified ratio of AR parameters

Expert Design System for Choosing the Optimum Size and Shape of Ultrasonic Transducer

— Estimation of the Elastic Dynamic Behaviour by Using the Structural Analysis Technique —

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1 Introduction

The purpose of this paper is to present details of an algorithm for performing the computer aided designing of ultrasonic transducer with a tool where the ultrasound energy is concentrated. The most typical and active applications of ultrasound energy are those such as ultrasonic working, ultrasonic machining and ultrasonic surgical operation. The length, the shape and the material's constants are very important factors in designing these ultrasonic instruments. The ultrasonic oscillation system is constructed in two units, that are an electronic oscillator and mechanical transducer unit which is mainly composed of two elements, i.e. magnetostrictive or piezo-electric transducer and a metallic horn. Although many reports have reported on the analyses of ultrasonic transducers, there are many unsolved problems concerning the practical application of these fundamental and essential studies to the improvement of technological process, especially in optimum designing for improving productivity.

This paper describes the structural analysis technique for the estimation of the resonance frequency and the dangerous position, and the technical application for performing the optimal strength design of ultrasonic oscillator composed of magnetostrictive or Langevin type transducers with varying cross section. These procedures are applied to carry out optimal design for high intensity ultrasonic surgery system having more complicated form.

2 Boundary Integral Formulation

As the diameter of the ultrasonic transducer used as the machining tool or the surgical knife is much smaller than the length, one dimensional analysis using the boundary element method is found to be the most efficient and rational approach for deriving the boundary values relationships. One of the important problems concerning the axial vibration is that of the case where the cross-sectional area varies in the axial direction. In the free vibration, when the distribution force is zero, the boundary-value problem for a thin rod in longitudinal vibration reduces to the differential equation

$$\frac{\partial}{\partial x} \left(EA \frac{\partial u}{\partial x} \right) = \rho A \frac{\partial^2 u}{\partial t^2} \quad (1)$$

We assume Young's modulus E is constant, and write for convenience $X = x/l$, $\omega_0 = c/l$, and $c = \sqrt{E/\rho}$. l is the length of rod element. Because free vibration is harmonic, we can write $u = U \exp(j\omega t)$ and eliminate the time dependence.

From the weighted residual statement, the following expression is obtained.

$$\begin{aligned} & \int_0^1 U^* \left[A \frac{d^2 U}{dX^2} + \frac{dA}{dX} \frac{dU}{dX} + A \left(\frac{\omega}{\omega_0} \right)^2 U \right] dX \\ &= U^* \frac{1}{Z_0 \omega_0} f \Big|_0^1 - \theta^* \frac{A}{A_L} U \Big|_0^1 + \frac{1}{A_L} \int_0^1 \frac{dU^*}{dX} \frac{dA}{dX} U dX \\ &+ \frac{A(\xi)}{A_L} U(\xi) = 0 \end{aligned} \quad (2)$$

where the weighting function U^* is the fundamental solution defined by;

$$\frac{d^2 U^*}{dX^2} + \Omega^2 U^* = \Delta(X - \xi) \quad (3)$$

ξ is the internal point subjected to a load of unit intensity, Ω is the frequency ratio ω/ω_0 and $\Delta(X - \xi)$ the Dirac delta function. The subscripts R and L refer to the boundary values of quantity on the right $X = 1$ and the left $X = 0$ sides respectively. Z_0 is the mechanical impedance $Z_0 = EA_L/c$. Equation.(2) includes a domain integral whose integrand consists of

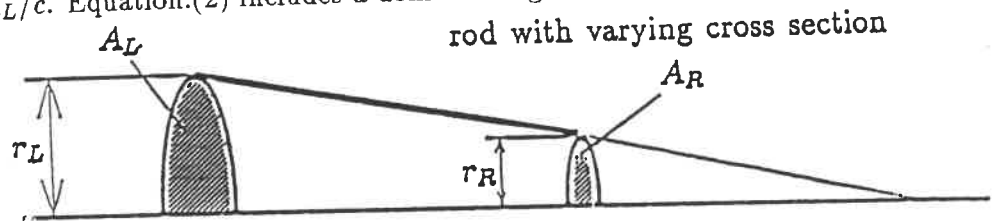


Fig.1 Axial vibration of frustum

Table.1 Ratio of frequency $\Omega = \omega/\omega_0$

left(r_L)	right(r_R)	r_L/r_R	exact	calc
1.15	1.15	1	1.5708	1.5707
1.25	1.15	1.09	1.6242	1.6242
1.5	1.15	1.30	1.7436	1.7432
1.75	1.15	1.52	1.8462	1.8453
2.0	1.15	1.74	1.9356	1.9340
2.25	1.15	1.96	2.0142	2.0117
2.75	1.15	2.39	2.1459	2.1422
3.25	1.15	2.83	2.2522	2.2469

Analyzed by Transfer Matrix Method

unknown variable U . So, the following approximation of the displacement at internal point X is used to evaluate the domain integral.

$$U = \left(\frac{A_L}{2A} \cos \Omega X \right) U_L + \left(\frac{A_R}{2A} \cos \Omega(1-X) \right) U_R + \left(\frac{A_L}{2Z_0\omega_0 A} \frac{\sin \Omega X}{\Omega} \right) f_L - \left(\frac{1}{2Z_0\omega_0} \frac{A_L \sin \Omega(1-X)}{A \Omega} \right) f_R \quad (4)$$

Here, f_L and f_R are axial forces acting on the left and right sides. Thus we have for boundary points $\xi = 0$ and $\xi = 1$

$$\begin{Bmatrix} f_R \\ U_R \end{Bmatrix} = \begin{bmatrix} a'_{11} & a'_{12} \\ a'_{21} & a'_{22} \end{bmatrix}^{-1} \begin{bmatrix} b'_{11} & b'_{12} \\ b'_{21} & b'_{22} \end{bmatrix} \begin{Bmatrix} f_L \\ U_L \end{Bmatrix} \quad (5)$$

$$a = \frac{1}{A_L} \frac{dA}{dX}, \quad a'_{11} = \frac{1}{2\Omega Z_0\omega_0} \int_0^1 a \frac{A_L}{A} \sin \Omega(1-X) \cos \Omega(1-X) dX, \text{ etc.}$$

The domain integral whose integrand consists entirely of known values can be evaluated numerically. For the case with uniform cross section rod, the evaluation of this integral is not necessary because a becomes zero.

Equation.(1) is written in the transfer matrix form relating the state vector on the left side to that on the side. If the nonuniform rod is divided up into elements, the frequency equation for any type of condition can be solved by a computer program.

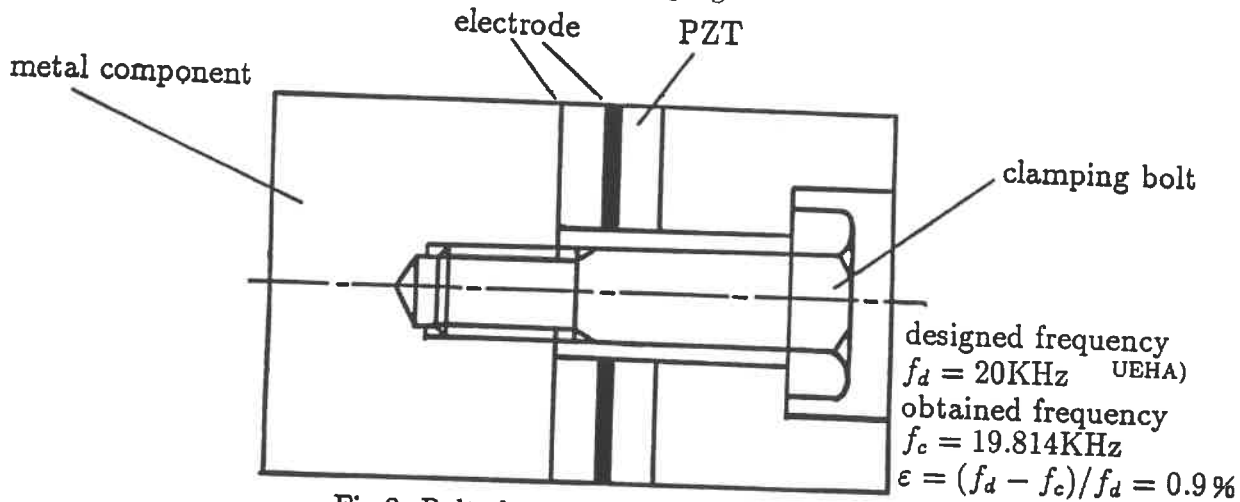


Fig.2 Bolt-clamped Langevin type transducer

Table.2 Relative ratio of frequency

	Mode.1	Mode.2	Mode.3
$(f_e - f_c)/f_e$	0.0090	0.0015	-0.0016

f_e : measured , f_c : calculated

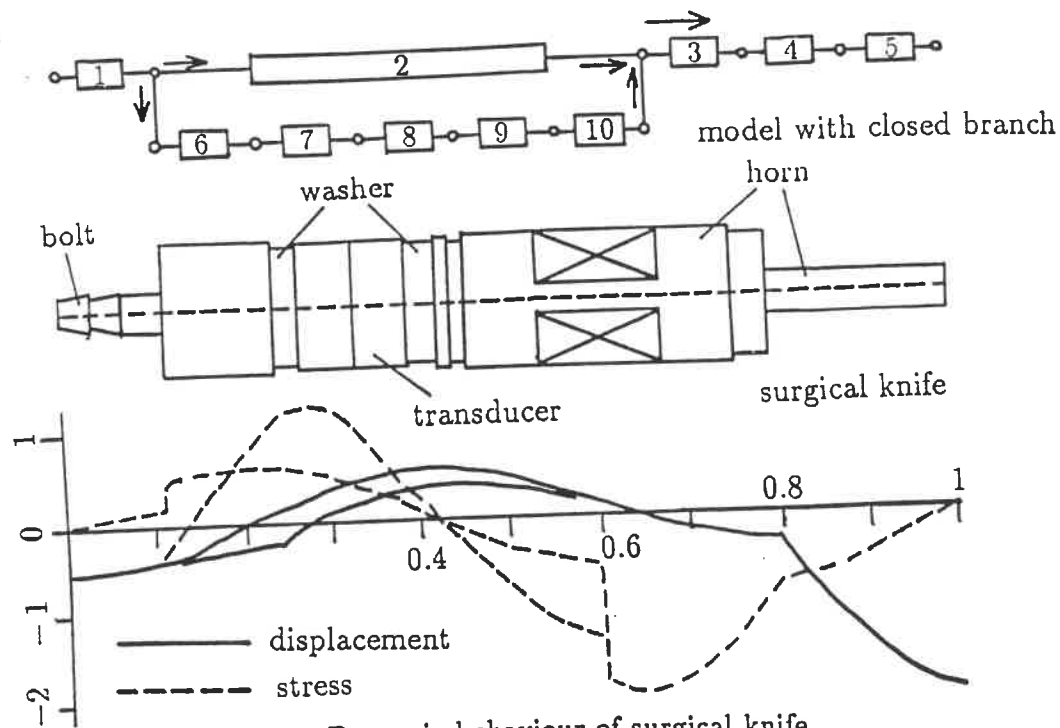


Fig.3 Dynamic behaviour of surgical knife

3 Expert System for Designing Surgical Knife

To check the validity of the formulation, we consider first a conical rod as shown in Fig.1, and divide it into 10 elements having same length. From the obtained results shown in Table.1, it is found that good accuracy can be obtained with small number of elements. The bolt clamped Langevin type transducer are also analyzed by using the program developed in this study. The resonance value of BLT transducer shown in Fig.2, the resonance frequencies obtained by this method are found to be successfully estimated with less than the error of 1 %.

The surgical operation is the another field that benefits from exploiting the high power of ultrasound. A special advantage of ultrasonic surgical knife for operation is that deeper-seated diseased tissues(tumours, cancers, calculi, etc) can be speedily and accurately destroyed and removed in the body of the patient without damaging unaffected tissues and veins surrounding the lesion. The transducer and horn are designed to amplify microscopic amplitude of oscillation for producing a very high terminal velocity at the sharp edge of the surgical knife. As the size and the physical characteristics of materials determine the resonant frequency, it becomes very important for designer to take account of the variations of the supplied material quality for determining the final size of his product. In order to avoid the undesired lateral vibration that blasts off the contamination and dissipates the power of cutting during the surgical operation, the amplitude of lateral vibration at the cutting end should also be examined. Tables.2 and Fig.3 show the numerical examples of resonant frequencies of a surgical knife. This method described here also presents a convenient and practical procedure which automatically calculates the length or the diameter of transducer satisfied with the design specifications such as applied frequency, resonant frequency or the mechanical properties of the material of transducer.

Model-based Automatic Generation of Control Sequence from Design Information

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Abstract

In this paper, we propose a new model-based technique to automatically generate sequence control software from design information of a design object. This technique solves the problem of software development of control sequence that is one of the bottlenecks in developing mechatronics machines. In this technique, by searching through the physical causalities of a design object represented by Qualitative Process Theory, the sequence control software, which realizes sequence of needed functions given by a designer, is generated. We also implemented an experimental system named Sequence Control Program (SCP) Generator based on this technique and developed an experimental photocopier that executes sequence control programs generated by the SCP Generator. The SCP Generator succeeded in generating a sequence control program that operates the photocopier correctly. The system also succeeded in activating a redundant function by modifying the sequence control program when the photocopier had a fault.

1 Introduction

It is well known that one of the bottle necks in developing mechatronics machines is software development of, among other things, sequence control. Although some researchers (*e.g.* [1, 2]) proposed techniques for automatically generating software, these techniques are not useful for generating a sequence control software. One of the reasons is that such techniques are not integrated with mechanical CAD. In this paper, we propose a new model-based technique to automatically generate sequence control software from design information of a design object.

2 Strategy for Sequence Control Software Generation

For the integration of a software generation system for sequence control with mechanical CAD, the designer's intentions and enough information and knowledge about the design object should be transferable between them. To realize it, we use FBS modeler [3] that we have developed as a prototype of an intelligent CAD for conceptual design and develop a technique that generates sequence control programs by using models of design objects in the FBS modeler (see Figure1). As shown in Figure1, designers

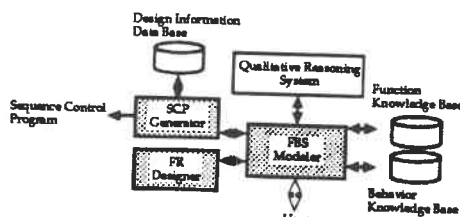


Figure 1: Architecture of the SCP Generator execute conceptual design on the FBS modeler. In the FBS modeler, a model of a design object (called a FBS model) preserves designers' intentions in the form of the functions and physical causalities based on Qualitative Process Theory (QPT)[4]. After conceptual design and parametric design are finished, the Sequence Control Program (SCP) Generator, which implements the technique proposed in this paper, generates a sequence control program that satisfies the designers' intentions represented as a sequence of needed functions in the FBS model.

By developing this architecture, we can integrate a software generation system for control sequence with mechanical CAD and, therefore, solve the bottleneck of software development in the mechatronics design.

2.1 Function-Behavior-State(FBS) Modeler

The FBS Modeler deals with knowledge about physical causalities, which is represented and reasoned based on QPT, as well as the function

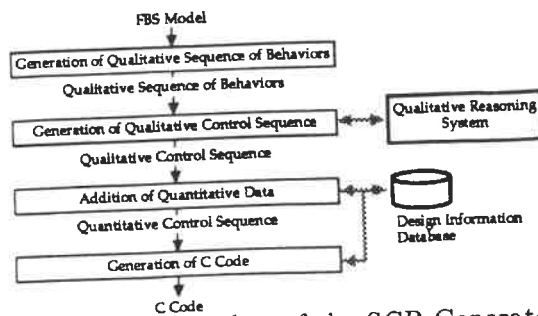


Figure 2: Algorithm of the SCP Generator

of the design object. Functions are represented in the scheme of function prototypes as shown in Table 1. *Decomposition* describes feasible candidates for detailing this function in the form of networks of subfunctions which include representation of needed temporal transitions among subfunctions. *F-B relationship* describes behaviors that can perform this function in the form of networks views based on QPT.

In QPT, a physical world is modeled with three components; *i.e.*, individuals, individual views, and processes. An individual represents an entity, such as a gear, water, etc. An individual view represents a state of an entity, such as velocity of a gear, temperature of water, etc. A process represents a physical phenomenon, such as rotating of a gear pair, boiling of water, etc. Therefore, the FBS Modeler represents behavior and state as a network of these concepts.

In this paper, we use a qualitative reasoning system [5] that implements QPT. In this system, individuals, individual views, and processes are represented in a uniform framework named a *view*.

3 Algorithm of Sequence Control Software Generation

In this section, we describe the algorithm of SCP generator. A sequence control program is generated from an input FBS model of the design object (see Figure2). In this figure, a *qualitative* control sequence means a sequence of states in a temporal order, while a *quantitative* one is the sequence in which time of each state is determined.

3.1 Generation of Qualitative Sequence of Behaviors

After inputting an FBS model, SCP Generator reasons out a sequence of behaviors those are

Table 1: Definition of a Functional Prototype

Item	Contents
Name	<i>verb + objectives</i>
Decomposition	subfunction networks
F-B Relationships	networks of views

Table 2: Sequence of Behavior

state	statel	state2
required function	function1	function2
views	phenomenon1 entity1	phenomenon2 entity1
initial parameter conditions*	S1 = low, S2 = low X = x1, V = 0 C1 = off, C2 = off C3 = off, C4 = off	

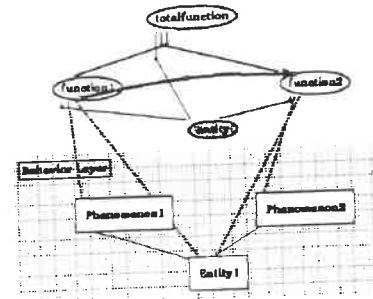


Figure 3: Example of the Input FBS model represented by physical views to satisfy a transition sequence of subfunctions in request.

Figure 3 shows an example of the initial FBS model. The sequence of behaviors, as shown in Table 2 is derived from the initial FBS model, because the needed functional sequence for realizing *total function* is *function1* → *function2* and *function1,2* are performed by activating *phenomenon1,2*, respectively, as shown in Figure 3.

3.2 Generation of Qualitative Control Sequence

At this stage, the system uses the qualitative reasoning system [5], which maintains consistency of behaviors, to reason out required parameter changes for the sequence of behaviors generated in the previous stage.

Firstly, the system creates a parameter transition map for activating views needed for the sequence of behavior. In the example with views defined in Table 3, the system creates a transition map shown in Figure 4. Table 3 denotes if the conditions of a phenomenon are satisfied,

*These conditions are described in each function by the designer.

Table 3: Definition of Each View

phenomenons	1	2	3	4
conditions	entity1 X=x1 S2=low C1= on	entity1 X=x3 S1=high S2=high C2=on	entity1 X=x2 S1=low S2=high C3=on	entity1 C4=on
influences	S2=high		S1=high	V=v0 $\frac{dX}{dt} = V$

states	initial State	state1	state2
occurring phenomena		phenomenon1	phenomenon2
sensors			
S1	low		high
S2	low	low	high
X	x1	x1	x3
V	0		
actuators			
C1	off	on	
C2	off		on
C3	off		
C4	off		

Figure 4: Specified Transition Map

states	initial State	state1	state3	state2
occurring phenomena		phenomenon1	phenomenon3	phenomenon2
sensors				
S1	low		low	high
S2	low	low	high	high
X	x1	x1	x2	high
V	0		v0	x3
actuators				
C1	off	on		
C2	off			on
C3	off		on	
C4	off	on		on

Figure 5: Modified Transition Map

its influences occur on the design object.

However, the map can be inconsistent; for example parameter S1 should be *high* at state2, although it is *low* in the initial state. Although some parameter values are controllable, other parameters should be controlled indirectly by activating appropriate views. In the example, the system adds a state named *state 3* to get phenomenon3 that changes the value of parameter S1 from *low* to *high*. Repeating this step, the system completes the consistent transition map shown in Figure 5.

3.3 Supplying Quantitative Data

At this stage, a quantitative sequence is generated by determining time intervals between states in the qualitative sequence. Since parametric changes according to time are determined by differential equations of parameters,

the system calculates the time intervals by integrating the equations. Namely, if α and β are related by equation $\frac{d\alpha}{dt} = \beta$ and β is constant between two states, the length of time Δt between the two states is calculated by equation $\Delta t = \frac{\Delta\alpha}{\beta}$. (If this kind of relations is not known, the system cannot calculate the time interval. But this is very rare if we have sufficient information.)

Then, the quantitative sequence is translated into control rules. In the example above, the following control rules are generated, where S1 and S2 are sensor values and C1, C2, C3, and C4 are actuator values.

```

if low(S1) & low(S2) & off(C1) & off(C2) &
off(C3) & off(C4)
then make(C1,on) & make(C4,on)
if low(S1) & high(S2) & on(C1) & off(C2) &
off(C3) & on(C4)
then after 100ms, make(C3,on)
if high(S1) & high(S2) & on(C1) & off(C2) &
on(C3) & on(C4)
then after 200ms, make(C2,on)

```

Finally, the system translates the rules into a C program.

3.4 Example

We implemented Sequence Control Program (SCP) Generator and developed an experimental photocopier which executes sequence control programs generated by the SCP Generator.

The hardcopy of the FBS model of the photocopier is shown in Figure 6. From this model, the SCP Generator succeeded in generating a sequence control program that operates the photocopier correctly (see Figure 7).

However, as shown in Figure 7 the position of the top of the output B is not correct, because the paper did not move as expected. This can be avoided by using sensor information. It is one of our future issues to incorporate sensor information into SCP Generator.

4 Application:

Self-Maintenance Machine

We have proposed "the Self-Maintenance Machine" that can maintain its functions flexibly even though faults happen in it [6]. In order to maintain its functions, we use functional redundancy, which is a design strategy for adding redundancies to a machine by using potential functions of its existing parts.

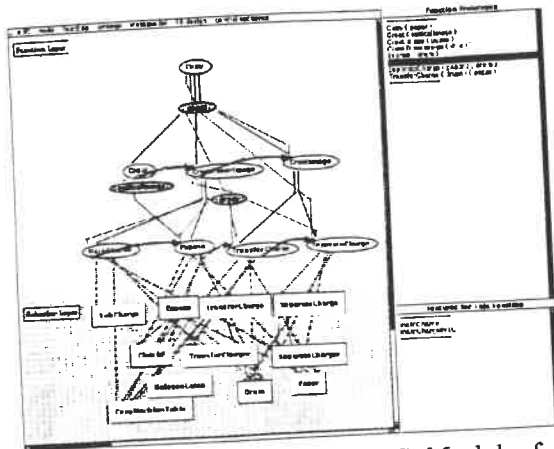


Figure 6: Hardcopy of the FBS Model of the experimental photocopier

Let us see an example of functional redundancy in a photocopier. Figure 8 shows the structure of a photocopier.

The function "to charge the drum" is performed by the main charger. FBS Modeler suggests that "to charge the drum" can also be performed by the transfer charger, of which the original function is "to transfer toner from the drum to the output paper," because the transfer charger's working principle is the same as that of the main charger. This means that the transfer charger has a potential function "to charge the drum."

When the main charger is broken, the SCP Generator succeeded in dynamically activating this redundant function (see Figure 9) by generating the control sequence that does not use the main charger.

The lower half of D is weaker than that of B, while the upper half of D is same as that of B. This is caused because the radius of the drum is too small, which is peculiar to this experimental photocopier.

5 Conclusions

In this paper, we proposed a method for model-based automatic generation of control sequence from design information. The SCP (Sequence Control Program) Generator, which implements this technique succeeded in generating a sequence control software automatically and, moreover, in making functional redundant machines dynamically adaptive against faults.

This method is applicable not only to a photocopier but also to mechatronic machines generally by replacing the knowledge base.

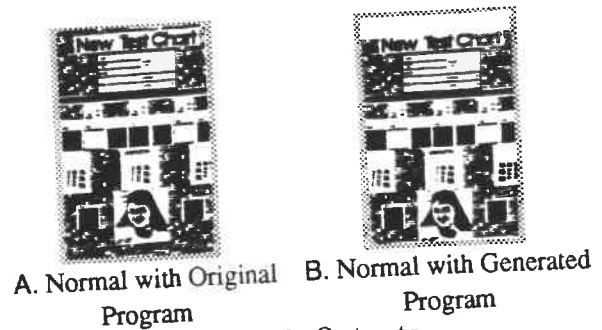


Figure 7: Outputs

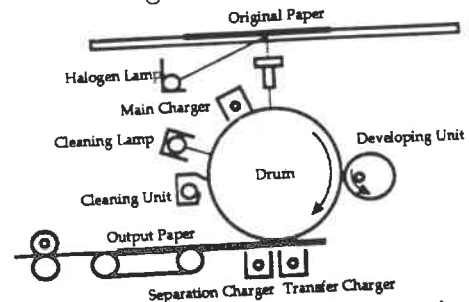


Figure 8: Section of a photocopier

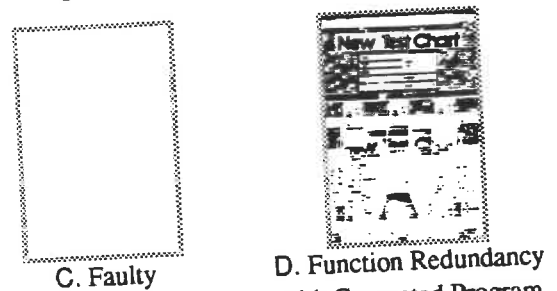


Figure 9: Outputs in Faulty States

Our future works includes to incorporate sensor information into SCP Generator.

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ACCURACY IMPROVEMENT OF MACHINE TOOL BY WORKPIECE-REFERRED CONTROL (2ND REPORT)

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1. Introduction

In ultraprecision machining technology, manufacture of more precise and larger elements is expected. In the case of turning, the machined form accuracy is determined by the so-called mother principle. Of course, it is important to improve the accuracy and stiffness of machine tool and to improve machining conditions. The author proposed a new system, called the workpiece-referred form accuracy control system (WORFAC), and confirmed an effectiveness of this manufacturing method[1]. In the previous report[2], two matters were found out: the effectiveness of simulation for estimating the formed accuracy, and the new function of the corrective servo by changing the feedback volume such that the error is multiplied by the corrective servo coefficient m ($0 \leq m \leq 1$). In this study, the effects of the coefficient m for disturbance elimination and for remained error convergency are cleared by both simulation on a computer and experiment of cylindrical turning.

2. Principle of the research

The Worfac is based on a concept that form accuracy is determined by the relative displacement between the workpiece and the cutting tool. Hence this method is a way of controlling the depth of micro-cutting via in-process measurement of the distance between the tool and the workpiece. The experimental system of cylindrical turning is illustrated in Fig.1. A micro-tool servo (MTS) and noncontact optical surface sensor (HIPOSS) are attached rigidly on

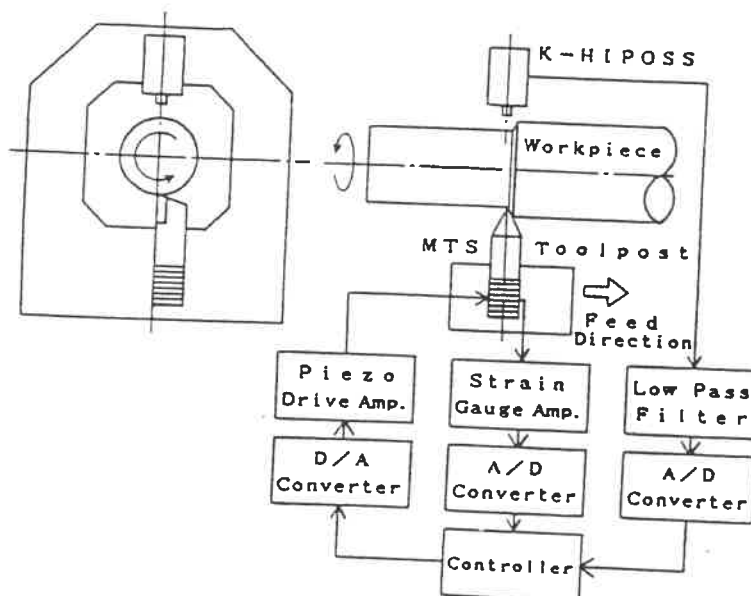


Fig.1 Block diagram of Worfac

the same post, and the workpiece is placed between the MTS and HIPOSS. The MTS consists of parallel leaf springs, a piezo-electric actuator, and strain gauges. Digital servo control is employed for changing the parameter of control freely.

3. Simulation of WORFAC

For simulation of the WORFAC on a computer, the mechanism of cylindrical turning is modeled and shown in Fig.2. The X-axis is defined as the direction of workpiece's axis, and x means the position of cutting point. The tool post motion to the radial direction of workpiece will be appeared as an error $T_0(x)$. In the case of servo-off turning, the amount of the displacement of the cutting tool and the sensor, as well as the cutting workpiece figure, are all same as $T_0(x)$.

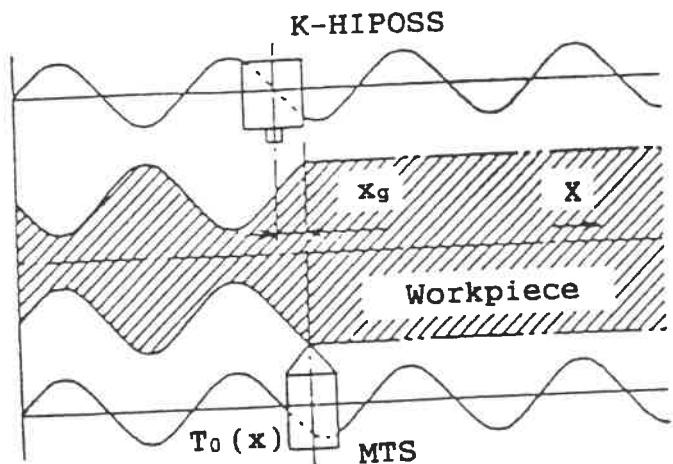


Fig.2 Simulation model of equipment

In the case of serovo-on turning, the motion of the tool and the figure change from $T_0(x)$ to $T(x)$ but the motion of sensor is still $T_0(x)$. If the axial distance between the cutting point and the sensing point is x_g , the sensing volume $S(x)$ is written by the equation (1).

$$S(x) = T_0(x) + T(x - x_g) + C \quad \dots \dots (1)$$

Where C means the off-set volume of the sensor in the radial direction and is set so that $S(x_0)$ becomes zero at the starting point x_0 of serovo-on turning. Here, using the advantage of digital control, the volume of feed-back to MTS is determined that which is $S(x)$ multiplied by a coefficient m (the corrective servo coefficient). Thus $T(x)$ is expressed by the equation (2).

$$T(x) = (1-m)T_0(x) - mT(x - x_g) - mC \quad \dots \dots (2)$$

4. Effects of coefficient m for disturbance elimination

The WORFAC system, making a short cut into the gap between the workpiece and the tool directly, can eliminate disturbance. In this study, the elimination of disturbance is investigated by changing the coefficient m . Figure 3 shows the results of simulation in the case of $m=0.5$ and $m=0.9$. Figure 4 is the relationship between the amplitude of artificially introduced disturbance A_d and that of eliminated disturbance. Referring to Fig.3 & 4, as the transitional response of the step disturbance, the $(1-m)A_d$ error causes. After converging, the $(1-m)/(1+m)A_d$ error is

remained. For the experiment of cylindrical turning, a CNC lathe and a diamond tool are used. The cutting conditions are as follows:

spindle speed 1000rpm,

$d=10\mu m$, $f=20\mu m/rev$.

The form accuracies are measured by a form tester, and the results are shown in Fig.5. They are the similar results as indicated by simulations.

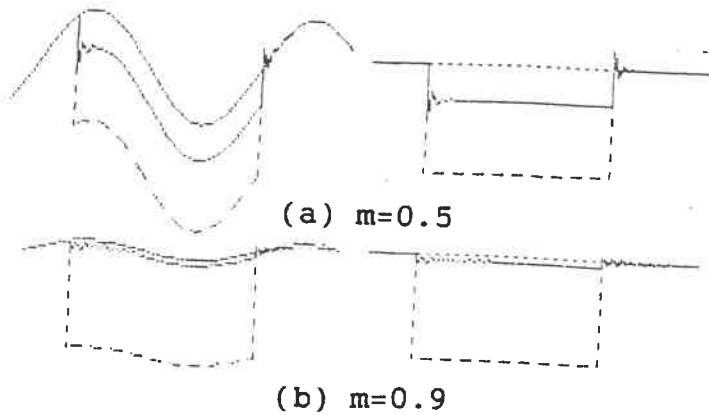


Fig.3 results of simulation

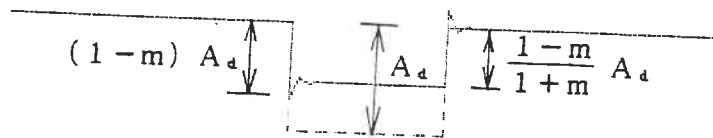


Fig.4 The degree of disturbance elimination for m

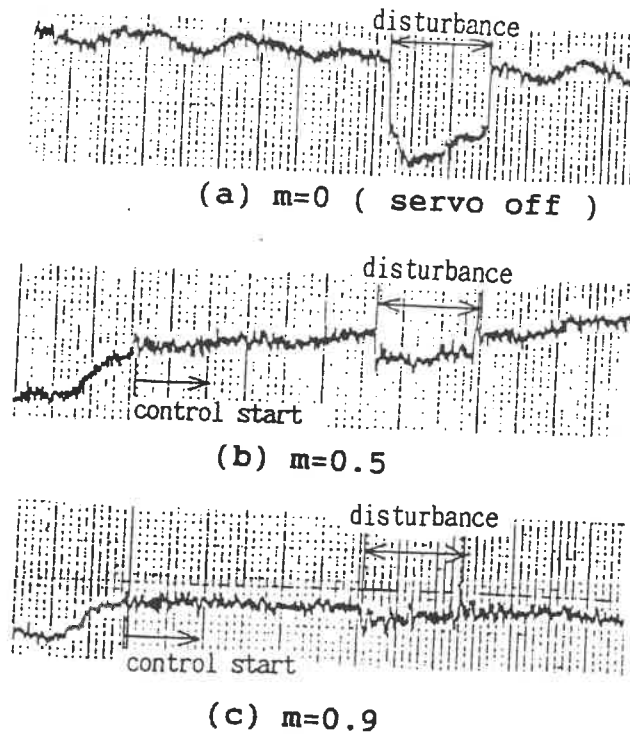


Fig.5 Results of cylindrical turning

5. Effects of coefficient m for remained error convergency

In the previous report, for making simple of the effects of the coefficient m , the axial distance x_g was regarded as very nearly zero. In this study, the convergency of remained error is

investigated by changing x_g . Figure 6 shows the results of simulation in the case of $x_g = \lambda / 6$ and $x_g = \lambda / 3$, where λ is the wave length of the slide error motion. Figure 7 is a diagram of the relationship between x_g / λ and M , where M is the ratio of form error at servo-on to that at servo-off. These relationships have periodicity on λ . If $x_g = n\lambda$ (n : positive whole number), M becomes $(1-m)/(1+m)$. In the case of $x_g = (n+1/2)\lambda$, M becomes 1 and the form error at servo-on is the same as that at servo-off.

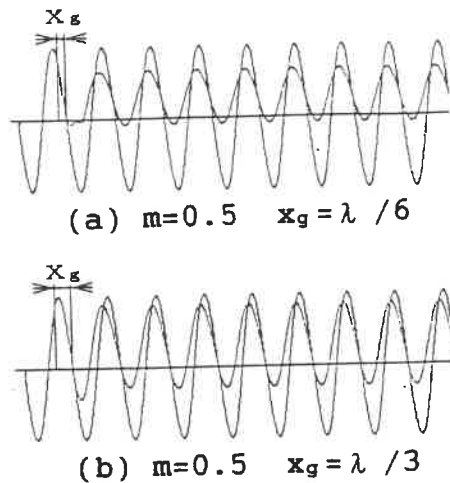


Fig.6 Results of simulations

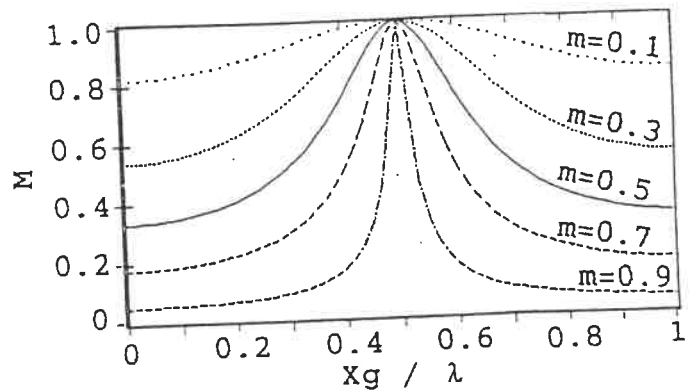


Fig.7 Diagram of relationship between x_g / λ and M

6. Conclusion

Workpiece-referred form accuracy control (WORFAC) is applied an ordinary NC machine tool, and a simulation for the situation are carried out on a computer. The results of both simulation and experiment of cylindrical turning are compared and investigated.

- (1) Effects for disturbance elimination
 - ① Due to m , the degree of disturbance elimination is changed.
 - ② As the transitional response of the step disturbance, the $(1-m)A_d$ error causes.
 - ③ After converging, the $(1-m)/(1+m)A_d$ error is remained.
- (2) Effects for remained error convergency
 - ① The amplitude of remained error has a periodicity of λ .
 - ② In the case of $x_g = n\lambda$ (n : positive whole number), M becomes $(1-m)/(1+m)$.
 - ③ In the case of $x_g = (n+1/2)\lambda$, M becomes 1 and the form error at servo-on is same as that at servo-off. This case is such a special point as uncontrollable for form accuracy improvement.

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AN INITIALIZATION PROCEDURE FOR CALIBRATION WITH A THREE DIMENSIONAL TRACKING SYSTEM

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ABSTRACT : In this paper an initialization procedure for a tracking system is discussed, the mathematical expressions involved are presented and some aspects related to the use of an instrument without absolute scales for displacement measurements are also discussed.

KEYWORDS : TRACKING SYSTEMS, MACHINE TOOLS CALIBRATION, ROBOT EVALUATION.

Introduction : With the ever increasing number of complex machines in the manufacturing industries, from time to time it becomes necessary to evaluate positioning accuracy of those machines.

There are many difficulties in performing an evaluation of machines with many degrees of freedom. It can be said that it is proportional to their complexity and it increases with the increase in the number of degrees of freedom of the machine. Two different general approaches can be used in the evaluation of complex machines: the internal calibration and the external calibration.

The internal calibration can be said to be the one that is concerned with the measurements of those joint variables, which define the machine position in the working volume. To perform a complete internal calibration, it is necessary to use extensive mathematical modelling. These models are derived for each individual type of machine what will permit one to analyse and quantify the effect of individual errors on the "volumetric error". The complete definition of the positioning accuracy of a machine can be performed through measurements of the individual errors associated with each degree of freedom, providing to the previously defined mathematical model reliable data.

The external calibration method is concerned with the measurement of the spatial location of the probing point, which directly provides information about the positioning capability of the system. To accomplish such a

task, it is necessary to use a measurement system capable of measuring three-dimensional positions.

In this paper an initialization procedure for a three-dimensional measuring system is presented, and the mathematical expressions used to perform the transformation of the coordinates are discussed.

The evaluation and the initialization procedures : As already mentioned, to evaluate the positioning accuracy of a complex machine, it is necessary to have a system capable of tracking a target point "P" in a three-dimensional space.

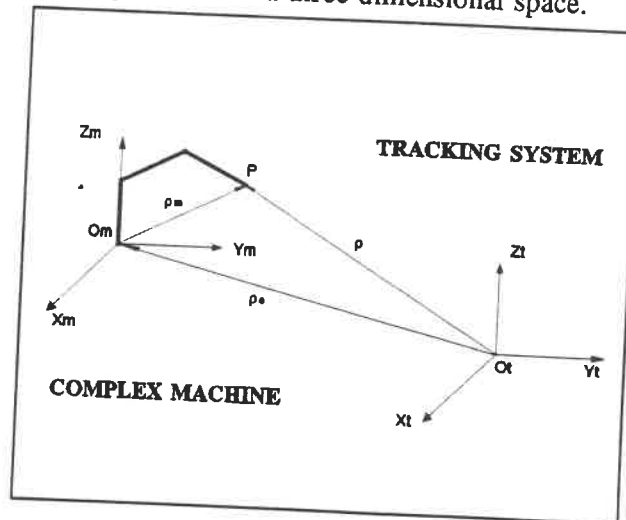


Figure 1. General configuration for external positioning evaluation. Complex machine (m), tracking system (t).

As shown in figure 1, an absolute coordinate reference frame $O_t X_t Y_t Z_t$ is fixed to the calibration

instrument and the frame $O_m X_m Y_m Z_m$ is fixed to the machine to be calibrated.

Before starting the calibration procedure, the relative position ρ_o should be determined, as well as the attitude of the machine with respect to the calibration system^[9].

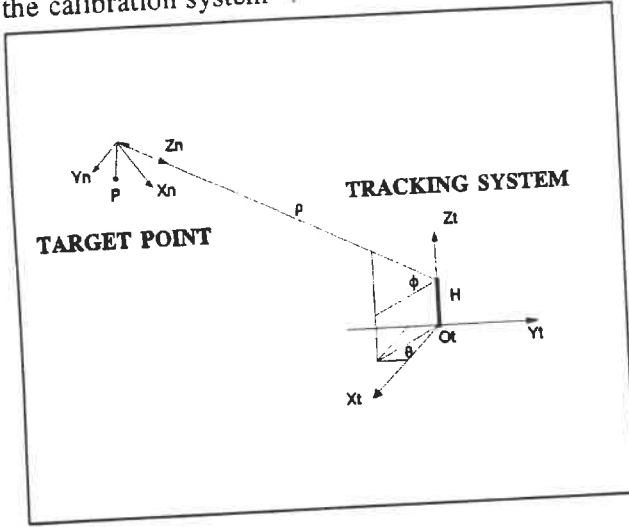


Figure 2 : The reference coordinate frame of the tracking instrument.

Knowing the translation vector ρ_o and the attitude matrix, the vectors ρ_m and ρ measured respectively by machine and by the calibration instrument, can be written to a common coordinate reference system. This will permit a direct comparison between the vectors, allowing the evaluation of the positioning accuracy of any point "P" located in the working volume of the machine.

Using a three-dimensional tracking system, the spatial position of a target can be defined as a vector $\rho = f(\rho_i, \phi_i, \theta_i)$, where ρ_i is the module of the vector, ϕ_i is the elevation angle and θ_i is the azimuth angle. All these spherical coordinates are referred to a previously established coordinate frame and can be easily written in a cartesian coordinate system $O_t X_t Y_t Z_t$, as shown in figure 2 and expressed as in equation 1.

$$\begin{bmatrix} x_t \\ y_t \\ z_t \\ 1 \end{bmatrix} = \begin{bmatrix} S(\theta) & -C(\theta)S(\phi) & C(\theta)C(\phi) & -\rho C(\theta)C(\phi) \\ -C(\theta) & -S(\theta)S(\phi) & S(\theta)C(\phi) & -\rho S(\theta)C(\phi) \\ 0 & C(\phi) & -S(\phi) & \rho S(\phi) + H \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_n \\ y_n \\ z_n \\ 1 \end{bmatrix} \quad (1)$$

$$x_t = -\rho C(\phi)C(\theta)$$

$$y_t = -\rho C(\phi)S(\theta)$$

$$z_t = \rho S(\phi)$$

(2)

The equation 2 represents a simplified form of the spherical to cartesian transformation.

Since there is no absolute scale for displacements measurements in the tracking system, an initialization procedure and an Euclidian geometric analysis must be performed to determine the value of the translation vector ρ_o . As shown in figure 3, by performing a set of sequential measurements, a group of angular coordinates (ϕ_i, θ_i) and variations $\Delta\rho_i$ can be obtained. Measurements of the points $P_1, P_2, \dots, P_4$ over each preferential axes must be performed, starting from the origin O_m of the machine reference frame. It will define an equation system that allows the calculation of the translation vector and the attitude of the machine.

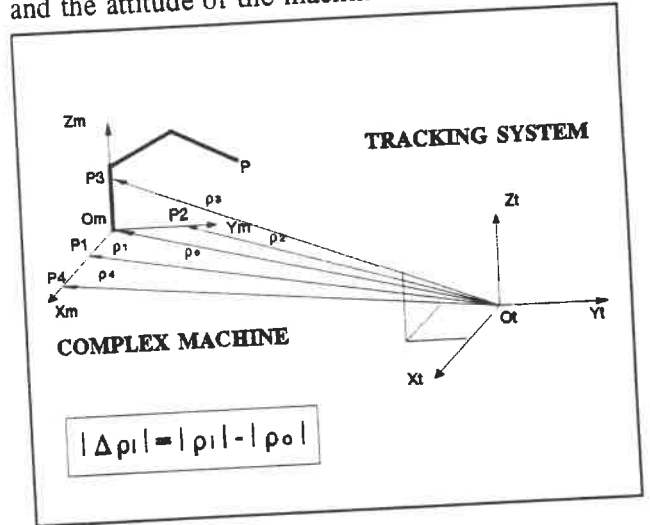


Figure 3 : The proposed configuration for the initialization procedure.

Defining $[V_i]_t$ as the direction vectors^[9] for the spherical-cartesian transformation, the absolute value of the translation vector ρ_o can be calculated by expression 3 that is written in a symbolic form for convenience of presentation. Since it is a vectorial relationship, it is impossible to perform the quotient as indicated here.

In the initialization procedure the values of Δx_i , Δy_i , Δz_i are the distances between each point P_i and the origin O_m on each preferential axes X_m , Y_m and Z_m of the machine. The calculation of ρ_o can be made using each component of the direction vectors. It is important to observe that, the way of solving the equation (3) has a direct relationship with the accuracy of the calculated value.

$$\rho_o = \frac{\left[\frac{\Delta \rho_1 [V_1]_t}{\Delta x_1} - \frac{\Delta \rho_4 [V_4]_t}{\Delta x_4} \right]}{\left[\frac{[V_4]_t}{\Delta x_4} + \frac{(\Delta x_4 - \Delta x_1)}{\Delta x_1 \Delta x_4} [V_0]_t + \frac{[V_1]_t}{\Delta x_1} \right]} \quad (3)$$

The attitude of the machine can also be readily calculated by the equations (4), (5) and (6), where $\{o\}$ is the orientation vector, $\{a\}$ is the approach vector and $\{n\}$ is the normal vector^[3].

$$\{n\} = \frac{\Delta \rho_1 [V_1]_t + \rho_o ([V_1]_t - [V_0]_t)}{\Delta x_1} \quad (4)$$

$$\{o\} = \frac{\Delta \rho_2 [V_2]_t + \rho_o ([V_2]_t - [V_0]_t)}{\Delta y_2} \quad (5)$$

$$\{a\} = \frac{\Delta \rho_3 [V_3]_t + \rho_o ([V_3]_t - [V_0]_t)}{\Delta z_3} \quad (6)$$

Once the initialization procedure is completed, the mathematical relationship between the coordinates of a point in the machine reference system and its coordinates in the instrument reference system can be expressed in matricial form as shown in the equation 7.

$$\begin{bmatrix} x_t \\ y_t \\ z_t \\ 1 \end{bmatrix} = \begin{bmatrix} \{n\} & \{o\} & \{a\} & \rho_o [V_0]_t \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x_m \\ y_m \\ z_m \\ 1 \end{bmatrix} \quad (7)$$

The error propagation : Observing the equationing presented here, one can ask about the effect of the uncertainties of the measured variables over the calculated results. This can be easily answered by the application of the rules of the error propagation analysis.

The importance of such a procedure, does

not only lie in the knowledge of the variations, but mainly in the diagnostic properties of the result. This is what permits one to foresee the variables that most influence the results, showing where special care must be addressed. Due to the complexity of the resulting expressions they are not presented here in full.

The cartesian coordinates obtained in the tracking system, are influenced by the uncertainties of the variables measured in the tracking head as shown in equation (8).

$$\begin{bmatrix} \delta x_t \\ \delta y_t \\ \delta z_t \end{bmatrix} = \begin{bmatrix} \rho C(\phi) S(\theta) & 0 & \rho C(\theta) S(\phi) & -C(\theta) C(\phi) \\ -\rho C(\theta) C(\phi) & 0 & \rho S(\theta) S(\phi) & -S(\theta) C(\phi) \\ 0 & 1 & C(\phi) & S(\phi) \end{bmatrix} \begin{bmatrix} \delta \theta \\ \delta H \\ \delta \phi \\ \delta \rho \end{bmatrix} \quad (8)$$

The influence of the variable "H" can be observed in the equation, which takes into account the effect of the error of axis of rotation in the elevation axis of the tracking system.

Variations for the equations (4), (5) and (6) are obtained through the Jacobian matrix^[3] $[J]$ as shown in expression (9).

$$\delta[A] = [J] \delta[V] \quad (9)$$

where

$$\delta[A] = [\delta n_x \ \delta n_y \ \delta n_z \ \delta o_x \ \delta o_y \ \delta o_z \ \delta a_x \ \delta a_y \ \delta a_z]^T$$

$$\delta[V] = [\delta \rho_0 \ \delta \theta_0 \ \delta \phi_0 \ \delta \theta_1 \ \delta \phi_1 \ \delta \theta_2 \ \dots$$

$$\dots \ \delta \phi_2 \ \delta \theta_3 \ \delta \phi_3 \ \delta \Delta \rho_1 \ \delta \Delta \rho_2 \ \delta \Delta \rho_3]^T$$

Conclusions : Based on the initialization procedure here presented, a complete mathematical model for the determination of the attitude and the translation vector between the coordinate frames of a complex machine and a tracking system has been derived. Consequently the homogeneous transformation matrix has also been written.

Experimental tests and computational simulation have been conducted, and the following conclusions have been reached:

The initialization procedure proves to be effective and easily to be carried out when applied either to a robot, machine center or a three-

dimensional measuring machine.

Computational simulations have indicated that the proposed initialization procedure can provide the attitude matrix with data whose numerical accuracies are better than 10^{-12} , indicating its effectiveness for most applications.

Experimental tests and numerical simulations also indicated that the effect of the accuracy of the calibrated machine in the initialization procedure is very small and represented in the propagation expressions by errors of second order.

The tests have also indicated that the accuracy of angular measurements of the tracking head have a strong effect in the calculation of ρ_0 , as shown in expression 3, suggesting that special care must be addressed to the angular measurements and to the implementation of this procedure.

Acknowledgments.

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A MATHEMATICAL MODEL FOR ON-LINE MEASUREMENT OF MICROCHANNEL GEOMETRY IN DIAMOND-MACHINING TECHNIQUE

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I. INTRODUCTION

The diamond-bit-machining technique is finding wide applications due to the ease of machining small and delicate workpieces with a high degree of surface finish and close tolerances. A typical example is the manufacture of micro heat exchangers by machining parallel micro flow channels on copper foils of 100 μm in thickness [1]. The micro channels, uniformly spaced along the foils, have either a rectangular or trapezoidal cross section with the depth and width being around 80 μm . Since the dimensional accuracy of each individual channel and the uniformity of channel spacing have a major influence on the performance of the final product, it is essential to implement a precise and effective inspection technique to monitor the channel geometry. Off-line metrological techniques [2] may be used for this purpose, but they are time-consuming and expensive due to the submicrometer range of dimensions encountered in the final product. This creates the need for an inspection technique that can monitor channel geometry on-line.

Previous investigations concerning on-line metrology in precision machining were mainly concerned with the tool positioning and tool wear [3]-[4]. The need for a technique to perform on-line measurement with respect to the workpiece has been mentioned [5], but not addressed so far. This is basically due to the complex mechanism of micro-machining that has to be accounted for when developing such a technique. Unlike conventional machining where the shear along the shear plane and friction at the rake face dominate, the micromachining process involves significant sliding along the tool flank due to the elastic recovery of the workpiece material and plowing due to the large effective rake produced by the tool edge radius at very small depths of cut [6]-[7]. Figs. 1A and 1B show the difference between these two basic machining processes schematically. In this paper, the mathematical background required for the hardware implementation of a proposed novel on-line workpiece cutting geometry inspection technique is presented. The developed models were evaluated by lab experiments and have been verified as applicable under realistic cutting conditions.

II. PRINCIPLE

In the precision machining of copper foils, the force experienced by the tool in the radial direction (F_r) is the vector sum of the feed force (F_f) and the thrusts due to cutting (F_{tc}), flank-work interaction (F_{fw}), and plowing (F_{lp}). F_f and F_{tc} have a definite relationship to the depth and width of the cut taken during machining, while the remaining components account for the energy dissipation not directly related to the metal removal. Thus the radial force (F_r) can be used as a direct measure of the cutting geometry. In Fig. 2, the experimental set up used to implement this principle is shown where a piezo sensor has been used to measure the force F_r .

III. MODELING OF CUTTING GEOMETRY

The cutting geometry involved is shown in Fig. 3, where the depth of cut is given by:

$$h = \frac{1}{f(a)} \left[\frac{1}{0.4 \theta_i} (F_r - F_f - F_{ip}) \sqrt{\frac{v^3 t}{k \rho c}} - f(b) \right] \quad (1)$$

with h as the depth of cut, v as the mean cutting speed, t as the undeformed chip thickness, θ_i as the tool-chip interface temperature, k as the thermal conductivity of copper, ρ as the density of copper, c as the specific heat of copper, and $f(a)$ and $f(b)$ as functions of the outer radius of the foil r wrapped on the mandrel, and the fixture angle ϕ which is a constant. The equation is valid for both trapezoidal and rectangular cross sections. The cutting width w is either the same as the tool width (in the rectangular case) or can be easily determined using trigonometrical relationships (trapezoidal case). Since the thrust force components due to flank-work interaction (F_{if}) and plowing (F_{ip}) in equation (1) can not be easily measured, they are simulated by the following equations:

$$F_{if} = A_1 [k_1 V_1 \cos(\gamma_1) - k_2 \sin(\gamma_1)] + 2 A_2 [V_2 k_3 \cos(\gamma_2) - k_4 \sin(\gamma_2)] + f(c) \quad (2)$$

and

$$F_{ip} = A_3 [k_2 \cos(\gamma_3) - V_3 k_1 \sin(\gamma_3)] + 2 A_4 [k_3 \cos(\gamma_4) - V_4 k_4 \sin(\gamma_4)] + f(d) \quad (3)$$

where k_1 through k_4 are constants accounting for the friction coefficients, workpiece hardness, and the area spread out due to shearing between the copper foil and the diamond-bit, A_1 and A_2 are areas of flank-work interaction, A_3 and A_4 are areas undergoing plowing, γ_1 through γ_4 are the clearance angles of the diamond-bit, V_1 through V_4 are the particle velocities created by the stress wave, and $f(c)$ and $f(d)$ are functions to account for the axial thrust. Equations (2) and (3) give expressions for the energy dissipation during the micromachining process that is not directly associated with the metal cutting.

The tool-chip interface temperature θ_i presented in equation (1) is a function of the mean temperature of shear plane θ_s , the ambient temperature θ_o , the mean temperature rise due to the rake friction θ_r and flank-work interaction θ_n , and the flank factor s . The term θ_i is further related to the specific energy expenditure due to shearing at the shear plane and friction at the rake face. It does not take into account the specific energy consumed due to sliding at the flank-work interface. This is considered by the introduced flank factor s , which is defined as the ratio of the tool-chip interface temperature due to shearing, rake friction, and flank-work rubbing to the tool-chip interface temperature without flank-work rubbing. For simplicity, the model for the tool-chip interface temperature has been simplified in this paper by taking estimated constant values for all the parameters. Efforts are now in progress to develop a comprehensive temperature model that accounts for the realistic changes in tool-chip interface temperature when the depth of cut changes.

IV. SIMULATION AND EXPERIMENTAL RESULTS

The models developed in the previous section was simulated by a specially designed computer code and the results obtained have been compared to those obtained through lab experiments. For the experiment, a piezoelectric sensor with a resolution of 0.44 mN and a sensitivity of 6.6 mV/gm has been used to measure the radial forces. As shown in Fig. 4, the radial force is plotted as a function of the depth of cut which has been varied from 1 μm to 50 μm in increments of 1 μm . Both the simulational and experimental data show an increase of the radial force as the depth of cut increases. This coincides with the fact that as the cutting tool is getting deeper into the workpiece, the contact area between the tool and the workpiece becomes larger, and hence the resistance encountered by the force sensor. The simulational data increase linearly as expressed by the linear relationship between the radial force and the depth of cut in equation (1). Error analysis has shown that the maximum relative error between the measured and simulated values was below 7%, as given in Fig. 5. Besides machine vibrations or other background noises, the error can be traced to the fact that the simulation has been conducted by assuming constant temperature conditions and the mechanical losses was neglected. The error values at 10 μm and 15 μm are presumably due to the extra tool and workpiece chatter which has caused a noticeable increase of the experimental values.

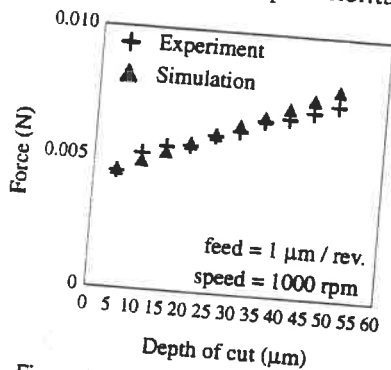


Figure 4: Plot of radial force Vs depth of cut.

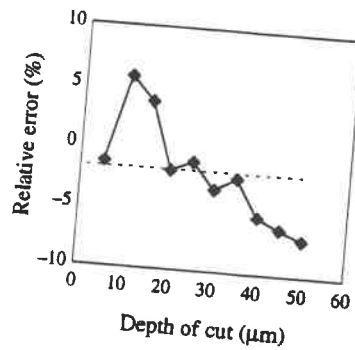


Figure 5: Plot of relative error Vs depth of cut.

V. CONCLUSION

By considering the physics of micromachining, this paper presents the mathematical background required for the hardware implementation of a novel cutting geometry inspection technique. The mathematical relations between the cutting geometry and the radial force, tool-chip interface temperature, and thrust due to the flank-work interaction and plowing have been derived and simulated. Also, equations have been developed for thrusts due to the flank-work interaction and plowing that account for the energy expenditure which is not dissipated through the cutting process. The proposed inspection technique can be used in the manufacture of micro compact heat exchangers to monitor on-line the geometrical variation of micro flow channels on thin copper foils as they are diamond-bit machined.

Experimental investigation have been conducted to evaluate the validity of the simulational results. Error analysis that has been performed to account for the deviation in experimental and simulational results shows that the maximum relative error is less than 7%. Efforts are in progress to expand and modify the models.

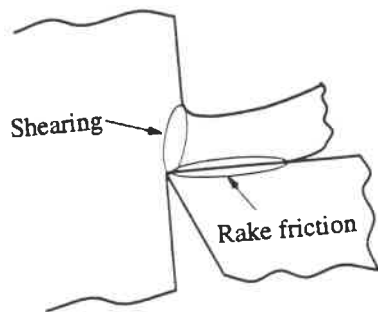


Figure 1A: Normal machining

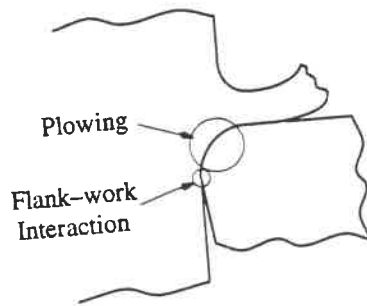


Figure 1B: Micro machining

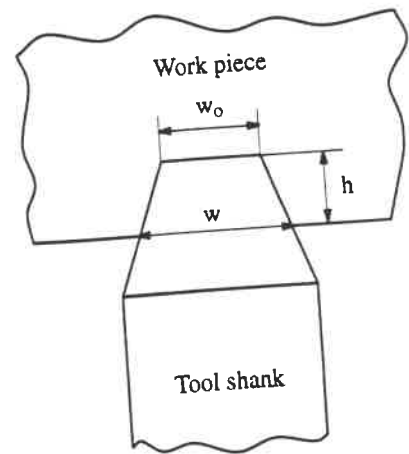


Figure 3: Cutting geometry

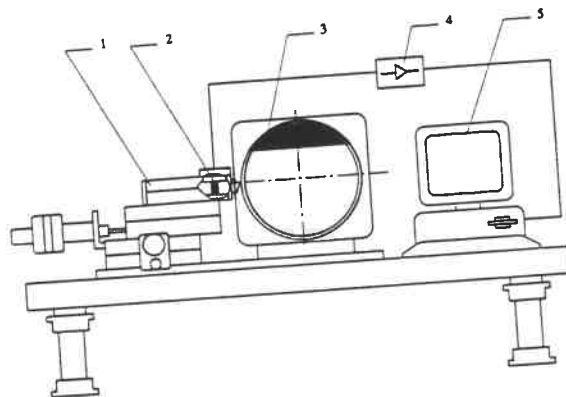


Figure 2: Set up used for implementing the on-line inspection technique

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THE ANALYSIS OF MOLECULAR SCALE ROUGHNESS EFFECT ON ADHESION AND FRICTION OF SOLIDS

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The optimal design of precision technology is not possible without prediction and consideration of surface forces in the contact clearance. The most significant for the Van-der-Waals surface forces are the clearance deviations in the nanometer range.

The authors have made an attempt to estimate the adhesional force and real area contact of the rough surface bond with an ideally smooth one without any additional idealizing suppositions on the shape and distribution of the asperities [1].

Model design scheme. The rough surface was specified by STM/AFM-image $z_i = Z(x_i, y_i)$. It was assumed that the bodies are elastically deformed. The deformed relief was represented as a layer of Winkler's springs which in the elastic case $P_i(\Delta z_i) = K \Delta z_i$ for the force and displacement ratio, where K - elasticity coefficient. The force of molecular inter-action through the clearance l_i was the Lennard-Jones potential $f(l_i)$ with parameters 3 and 9. Out of forces balance on each elementary area

$$P_i(\Delta z_i) + f(l_i + \Delta z_i) = 0$$

an additional approach of the surfaces Δz_i is determined due to noncontact adhesion. Then we proceed from initial surface to the changed one - z'_i for which $z'_i = z_i + \Delta z_i$ and receive distribution of forces on the contact. Summing up according of the image points and maintaining the equilibrium condition

$$F_S(d_0) - P(d_0) + N = 0$$

we may determine the integral characteristics of the contact: surfaces approach - d_0 ; real contact area A_r ; adhesion and friction forces depending on the external load N .

The suggested model was verified for the known theoretical solution with the sphere-plane geometry. The comparison of the calculations to the Johnson-Kendall-Roberts theory [2] and Derjagin-Muller-Toprov one [3] did not show inconsistency of the model.

Analyzing the results of the test-example the following can be noted:

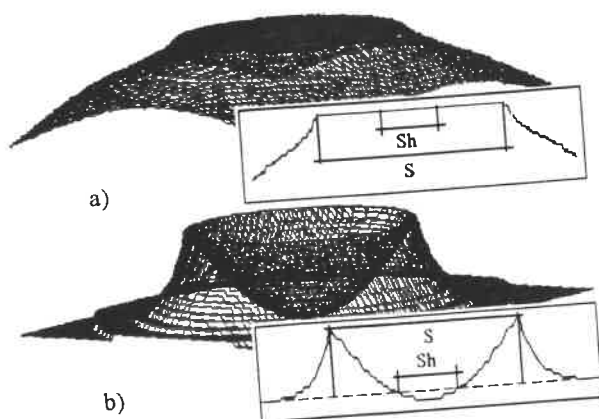


Fig. 1.

1) When there are surface forces acting through the clearance the contact causes expansion of the spot S as compared with the Hertzian one S_h (Fig. 1.a). Near the contact spot perimeter the profile of the sphere forms a transition necking.

2) The profile of distribution of forces is compressing (the negative value) within the

spot of soft crumpling of the material corresponding to the Hertzian and stretching beyond (Fig. 1.b). The adhesive attracting force increases towards the periphery of the ring increasing the contact spot S and diminishes outside it. There is some difference between the compressing portion of the of forces and the Hertzian pressure profile attributed to the specifics of the force approach concept used in the model.

The advantages of the suggested approach allow to waive a number of previous idealizing assumptions. Without any extra problem, its applicability to a real contact with surfaces of complex relief is most important. Application of the technique to the results obtained with the help of the scanning probing microscopy allows to investigate the adhesive and frictional processes of formation and separation of the contact at the molecular scale level.

Experimental measurements. The topographic study of the surfaces for the present paper was performed with the help of the scanning probe microscope with main characteristics: STM-axial sensitivity Z -0.01 nm, resolution the XY-plane scanning - 0.2 nm, maximal scanning area $8 \times 8 \mu\text{m}^2$; AFM - vertical and horizontal resolutions were 0.2 and 2 nm correspondingly, maximal scanning area $8 \times 8 \mu\text{m}^2$. The AFM include a device with a step not exceed $3 \mu\text{m}$ along the X and Y axes.

Since only some numerical estimates of calculated values can be mentioned at this stage of modeling, their mean tabulated values will be used as the specific surface energy together with the modules of elasticity of the studied materials.

The results of modeling and their discussion.

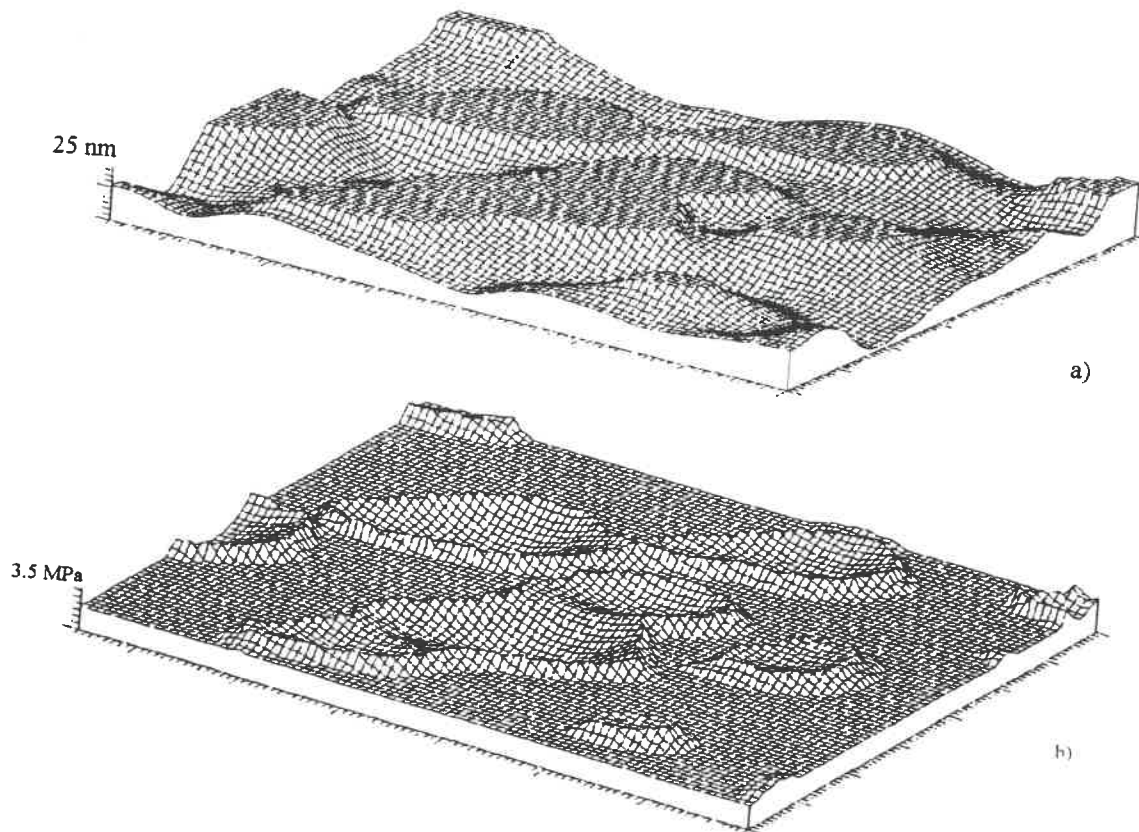


Fig. 2.

Fig. 2 represents the results of visualization of the process of approach of contacting surfaces and formation of the contact land (a) as well as distribution of the force field within the nominal contact zone (b). Alterations of the surface occur not only in the immediate contact points but also beyond the contact spots. As if the whole surface were attracted to the flat counterbody.

Since the deformation process through adhesive attraction involves additional bodies of the material, deformation is inhibited to some extent as well as the contact area, i. e., the surface layer undergoes some sort of hardening or "quasihardening". The effect gains significance as softer materials come into contact.

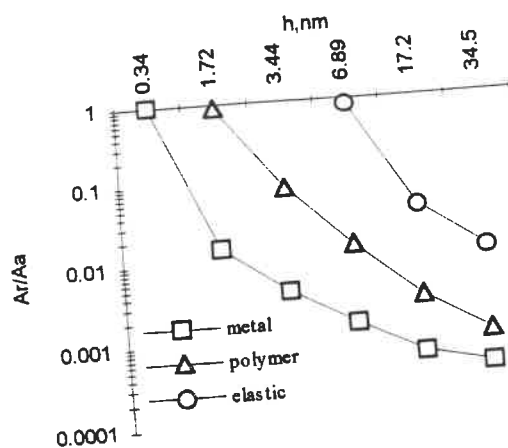


Fig. 3.

An important factor in forming the adhesive contact is the magnitude of deviation of the height of the surface. Fig. 3 shows the results of modeling the contact of different classes of materials when the range of spread of the heights of the surfaces is varied. It was assumed that the external compressive load was absent and only the forces of adhesion and resistance of the material counterbalanced the contacting system.

This relationship between the A_r (A_a -nominal area contact) and the spread of heights h indicates that the contact between highly elastic materials becomes continuous already at ~ 10 nm. This threshold is about 1--3 nm for polymers and it is less than 1 nm for metals.

Since the friction force is proportional to A_r , the A_r/A_a quantity under zero load may be considered as friction parameter of topography. Fig. 4 is the A_r/A_a diagram for a different surfaces (physico-mechanical properties of materials are assumed to be constant): (1a) texture substrate of hard Magnetic Disc (MD); (2a) MD with protective coating; (1b,2b) the same surfaces covered by gold. Also, molecular scale topography structure of materials is important factor of the friction contact.

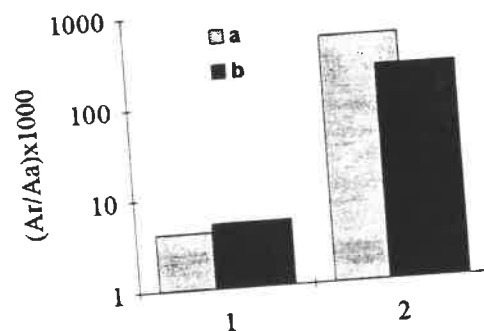


Fig. 4.

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SIMULATION OF VIBRATION IN A LARGE-SCALE ELECTRON MICROSCOPE

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1. Introduction

Electron microscopes can easily achieve magnifications of a few ten thousand times. However, they cannot observe thick specimens, because the wavelength of the electron beam is very short. The only way to improve electron beam penetration is to accelerate the electrons. This has led to the development of high-voltage microscopes in the 1 - 3 MV range. Meanwhile, the resolution of electron microscopes can be greatly improved by electron holography.^{1,2)}

Holographic electron microscopes require superior interference characteristics to conventional electron microscopes. In high-voltage electron holography, the most important problem is vibration. 1 MV high-voltage electron microscope has a structure in which the electron beam (abbreviated as EB) gun and the EB column are divided by a base frame, which is supported by rubber dampers on a supporting structure. The relative displacements between the EB gun and the base, and between the EB column and the base are currently of the order of 100 nm. However, these must be reduced to around 3 to 5 nm, the same as the diameter of the electron beam, in order to get a high-interference electron beam.

Previous vibrational studies of large-scale electron microscopes used models with one or two degrees-of-freedom, and only a single vibrational frequency of 4.4 Hz applied through the floor.^{3,4)} The present study, on the other hand, models the electron microscope as a lumped parameter system with six degrees-of-freedom relating to its main components. Our model employs a transfer function method, which allows it to cover a wide range of frequencies from 0 to 20 Hz of the main floor vibration.

2. Configuration and vibration of a large-scale electron microscope

2.1 Configuration of the microscope

The 1 MV microscope that we investigated is shown in Fig. 1. The base (vibrational displacement is shown by x_2) is between the upper and lower parts of the microscope. It rests on a supporting structure, which stands on the floor (x_0). There are four dampers between the base and the support (x_1). A high-pressure tank, which holds the EB gun (x_5), stands on the base. A hanging structure (x_3) consisting of four poles and a beam-control coil (x_6) are hung from the base. The observation chamber rests on the hanging structure and the EB column (x_4) is set on it. The microscope is 6.6 m high and weighs 13.8 tons.

2.2 Horizontal vibration of the floor and the microscope

The horizontal vibrations of the floor and elements of the microscope were measured.

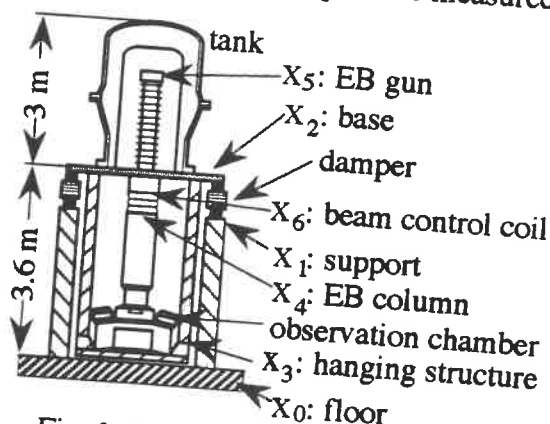


Fig. 1 1MV electron microscope. The variables X_1, X_2 etc. indicate horizontal vibrational displacements.

Maximum displacement of the floor was $x_0=159$ nm. Maximum displacements induced by the floor vibration were $x_1=254$ nm, $x_2=546$ nm, $x_3=363$ nm, $x_4=466$ nm, and $x_6=640$ nm. The maximum displacement of the relative horizontal vibration between the EB column and the base was 154 nm.

Measured values of the Fourier spectrum of the floor vibration are shown in Fig. 2. The main frequencies of the floor vibration lie between 0.6 Hz and 4 Hz. Vibrations larger than 20 Hz are negligible. Measured values of the Fourier spectrum of the support and the base are shown in Fig. 3. In the support, the measured value has peaks at 2 or 3 Hz and at 17 Hz. On the other hand, the base has a large peak at 1 Hz and small peaks at 2.5, 13, and 17 Hz.

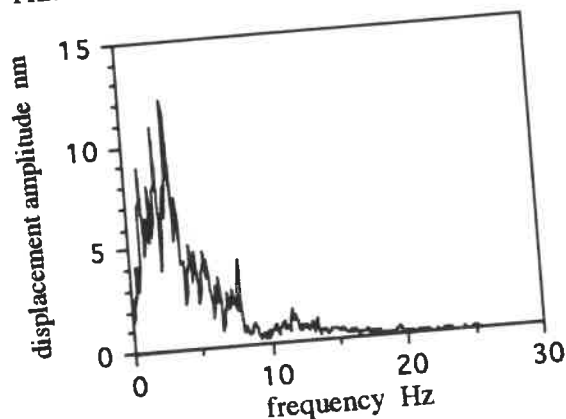


Fig. 2 Fourier spectrum of floor vibration (measured)

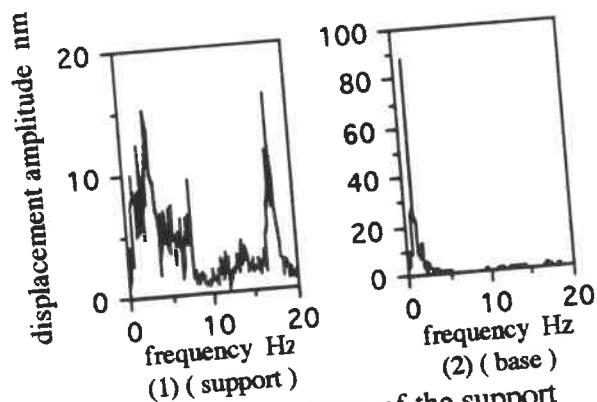


Fig. 3 Fourier spectrum of the support and the base (measured)

3. Vibration model for a large-scale electron microscope

3.1 Vibration elements and their constants

The vibration of the microscope was analyzed with a vibration model of a lumped parameter system. The vibration elements are (1) support, (2) base (including the tank), (3) hanging structure, (4) EB damper, (5) EB gun, and (6) beam-control coil. An easily visualizable vibration model for the microscope is shown in Fig. 4. The constants of the vibration elements are listed in Table 1.

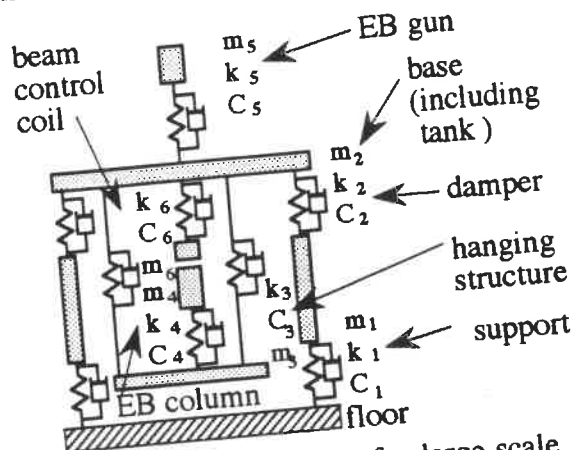


Fig. 4 Vibration model of a large-scale electron microscope

Table 1 Constants for each element of the vibration model

	mass kg	spring constant N/m	damping constant Ns/m
support	$m_1=2500$	$k_1=2.9 \times 10^7$	$C_1=2700$
damper and base	$m_2=8150$	$k_2=5.2 \times 10^5$	$C_2=8500$
hanging structure	$m_3=3750$	$k_3=7.0 \times 10^5$	$C_3=3100$
EB column	$m_4=1720$	$k_4=6.2 \times 10^7$	$C_4=3300$
EB gun	$m_5=100$	$k_5=4.8 \times 10^5$	$C_5=200$
beam control coil	$m_6=150$	$k_6=1.2 \times 10^8$	$C_6=1300$

3.2 Equations of motion

Six equations were obtained for the equations of motion of the elements of the microscope. Here s is the Laplace variable, $X_i(s)$ is the Laplace transform of x_i , and initial values are 0.

$$(m_6 s^2 + C_6 s + k_6) X_6(s) = (C_6 s + k_6) X_2(s) \quad (1)$$

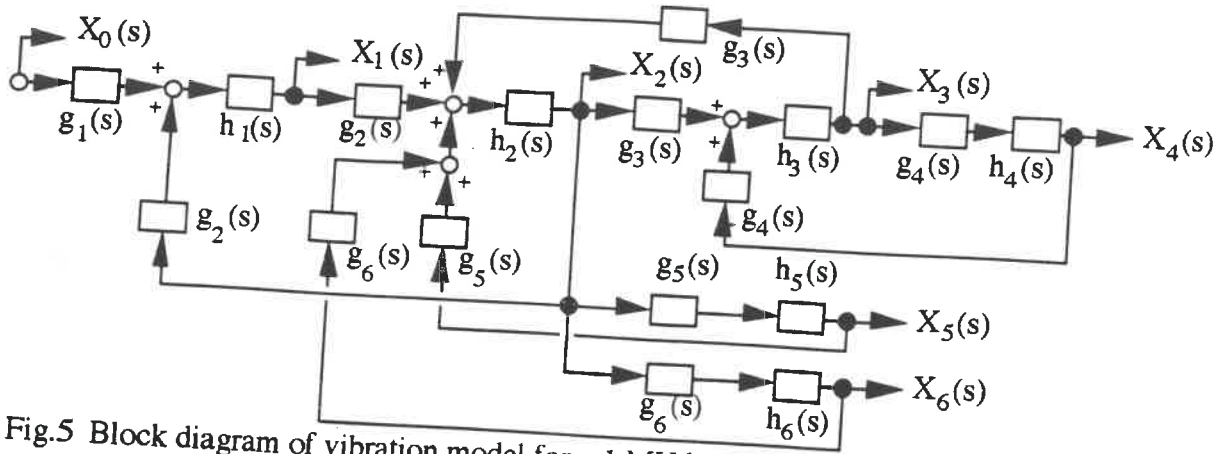


Fig.5 Block diagram of vibration model for a 1 MV large-scale electron microscope

$$(m_5 s^2 + C_5 s + k_5) X_5(s) = (C_5 s + k_5) X_2(s) \quad (2)$$

$$(m_4 s^2 + C_4 s + k_4) X_4(s) = (C_4 s + k_4) X_3(s) \quad (3)$$

$$\{m_3 s^2 + (C_3 + C_4) s + (k_3 + k_4)\} X_3(s) = (C_3 s + k_3) X_2(s) + (C_4 s + k_4) X_4(s) \quad (4)$$

$$\{m_2 s^2 + (C_2 + C_3 + C_5 + C_6) s + (k_2 + k_3 + k_5 + k_6)\} X_2(s) = (C_2 s + k_2) X_1(s) + (C_3 s + k_3) X_3(s) + (C_5 s + k_5) X_5(s) + (C_6 s + k_6) X_6(s) \quad (5)$$

$$\{m_1 s^2 + (C_1 + C_2) s + (k_1 + k_2)\} X_1(s) = (C_1 s + k_1) X_0(s) + (C_2 s + k_2) X_2(s) \quad (6)$$

3.3 Block diagram of the vibration model

Equations (1) to (6) are represented by the block diagram in Fig. 5. The transfer functions are :

$$g_1(s) = C_1 s + k_1 \quad (7)$$

$$g_2(s) = C_2 s + k_2 \quad (8)$$

$$g_3(s) = C_3 s + k_3 \quad (9)$$

$$g_4(s) = C_4 s + k_4 \quad (10)$$

$$g_5(s) = C_5 s + k_5 \quad (11)$$

$$g_6(s) = C_6 s + k_6 \quad (12)$$

$$h_1(s) = 1/\{m_1 s^2 + (C_1 + C_2) s + (k_1 + k_2)\} \quad (13)$$

$$h_2(s) = 1/\{m_2 s^2 + (C_2 + C_3 + C_5 + C_6) s + (k_2 + k_3 + k_5 + k_6)\} \quad (14)$$

$$h_3(s) = 1/\{m_3 s^2 + (C_3 + C_4) s + (k_3 + k_4)\} \quad (15)$$

$$h_4(s) = 1/\{m_4 s^2 + C_4 s + k_4\} \quad (16)$$

$$h_5(s) = 1/\{m_5 s^2 + C_5 s + k_5\} \quad (17)$$

$$h_6(s) = 1/\{m_6 s^2 + C_6 s + k_6\} \quad (18)$$

4. Comparison of simulation and experimental results

4.1 Fourier spectrum

The Fourier amplitude of each element is $X_i(j\omega) = T_i X_0(j\omega)$. (19)

Here, T_i is a transmissibility of floor displacement to each element. The calculated values of the Fourier spectrum of the support and the base are shown in Fig. 6.

(1) Support: The measured and calculated values have peaks at 2 or 3 Hz and at 17.4 Hz. The measured values are as large as 6%, but the forms of the two spectra are the same.

(2) Base: Both spectra have a large peak at 1 Hz, 88 nm for the measurements and 77 nm for the calculated values with an error of 14%.

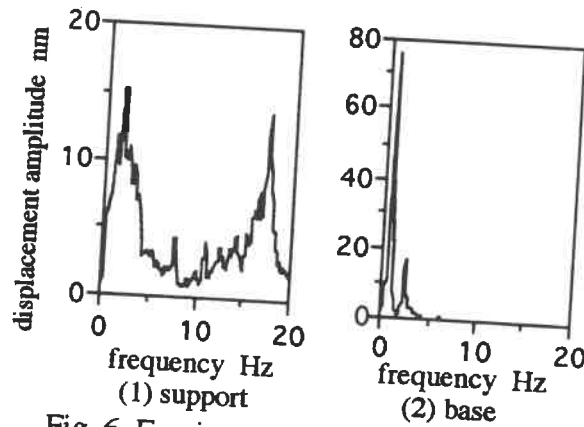


Fig. 6 Fourier spectrum of the support and the base (calculated)

4.2 Calculation of the maximum displacement

The displacements of each element in the time domain are obtained by reverse Fourier

transformation, but we are only concerned with the maximum value of the displacement, not its behavior over time. In the reverse Fourier transformation of $x_i(t)$, $\exp(j\omega_n t)$ does not exceed 1. Therefore, the direct integral of the Fourier spectrum is considered to be

$$x_{iM} = \frac{1}{2\pi} \sum_{n=-\infty}^{\infty} x_i(j\omega_n) \Delta \omega_n \quad (20)$$

This value can approximate the maximum displacement, because it is not lower than the maximum displacement. The direct integrals of Fourier spectrum (A) and the measured values of the maximum displacement (B) are shown in Table 2. The ratios of A to B in the support and the hanging structure are 2.36 and 1.52 respectively. Their approximations are not good. But the error in the base is 11%. The errors in the EB column, and beam-control coil range from 24% to 34%. The error of the displacement between the EB column and the base is 11%. Thus, it is possible to simulate the relative displacement to within 52% error except the support.

Table 2 Comparison of direct integral of Fourier spectrum and measured maximum displacement amplitude

	direct integral of Fourier spectrum (A)	maximum displacement amplitude (B)	A/B
x_1 : support	600 nm	254 nm	2.36
x_2 : base	490 nm	546 nm	0.89
x_3 : hanging structure	553 nm	363 nm	1.52
x_4 : column	627 nm	466 nm	1.34
x_6 : beam control coil	490 nm	640 nm	0.76
$x_4 - x_2$	137 nm	154 nm	0.89

5. Reducing vibration in the large-scale electron microscope

Vibration frequencies from the floor range from 0.6 to 4 Hz. The vibration of the floor invades the damper with amplification, because the resonant frequency of the support is high. Therefore, the following two measures should reduce the vibration in the

microscope.

(1) First measure: Use a monolithic support structure that integrates the floor part and column. For concrete, the mass should be about 2×10^5 kg. The rigidity (2×10^6 N/m) required for the supporting dampers can easily be attained. The resonant frequency of this system would be about 0.5 Hz ($k_1/m_1 = 10/s^2$).

(2) Second measure: Use a shell structure which fixes the four poles of the hanging structure with plates. The resonant frequency of the hanging structure (now 2.5 Hz) would be increased to 13 or 14 Hz.

The relative displacement between the EB column and the base is reduced to 85 nm from 137 nm after the first measure, but it is not sufficient. The addition of the second measure reduces the relative displacement to 2.8 nm, which is within the acceptable value of 5 nm. The relative displacement between the EB gun and the base is reduced to 1.9 nm from 35 nm after the first measure, but is not affected by the second. Thus we can meet the acceptable value of 3 nm if the error is 52%.

6. Conclusion

A vibration model has been developed for a 1 MV electron microscope. This model, which is a six degrees-of-freedom lumped parameter system, employs the transfer function method. It gives the vibration characteristics of the 1 MV electron microscope to within 52% error except for the displacement of the support. This simulation shows that the vibration of a large-scale electron microscope can be reduced to a level suitable for the high resolution electron microscopy used in electron holography.

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Finite Element Simulation of Three-Dimensional Continuous Chip Formation Processes

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Nomenclature

$A(\theta)$	constant in eq. (4)	$\dot{u}, \dot{v}, \dot{w}$	velocities in rectangular Cartesian coordinates
C	specific heat	V	cutting speed
C_1, C_2	wear characteristic constants	x, y, z	spatial fixed rectangular Cartesian coordinates
d	depth of cut	α^*	elastic-plastic parameter
dW/dL	wear volume per unit area and unit sliding distance	$\bar{\epsilon}$	equivalent plastic strain
E	Young's modulus	$\dot{\bar{\epsilon}}$	equivalent plastic strain rate
$\dot{\epsilon}_{kl}^{\text{mod}}$	modified rate of Lagrange strain	θ	temperature
f	feed rate	θ_s	specified temperature
G	shear modulus	θ_o	circumferential temperature
H'	work hardening rate	λ	friction characteristic constant
h	coefficient of heat transfer	ν	Poisson's ratio
K	thermal conductivity	ρ	mass density
k	constant in eq. (4)	$\bar{\sigma}$	equivalent stress
$M(\theta)$	constant in eq. (4)	σ_{ij}	Euler stress
m	constant in eq. (4)	σ_{ij}^*	Jaumann derivative of Euler stress
$N(\theta)$	constant in eq. (4)	σ'_{ij}	deviatoric component of Euler stress
n	outward normal	σ_t	normal stress on tool face
n'	constant in eq. (5)	τ_e	shear flow stress
Q	internal heat generation	τ_t	frictional stress on tool face
q	frictional heat flux		
t	time		

1 Introduction

Practical operations in metal machining are mostly performed under three-dimensional situations, where the workpiece is separated into a chip and finished surface by the front and side edges and the corner of a tool as well. Since the deformation depends on the three-dimensional shape of the cutting edges, there are many phenomena that should be solved as a three-dimensional problem, such as chip curl, tool fracture, boundary wear and burr formation.

The finite element method is an effective tool for the purpose of simulating the phenomena through visualization of the numerical information. The analysis requires a huge amount of calculation and costs much because it includes material nonlinearity, large deformation and deformation-temperature coupling, so that a two-dimensional approximation has been used at most.

However, it seems that developments in computing ability and software environment make it possible to perform the full three-dimensional finite element analysis. One can look forward to machining simulation under a virtual production environment being an inevitable procedure, aiming at optimising the process and at omitting a dry run accompanied by a waste of materials and time [1].

The purpose of the present study is to expand the machining simulation system from the existing two-dimensional one to a three-dimensional one which can be employed to predict cutting forces, chip formation, temperature and tool wear in a real turning process.

2 Theory

The metal machining process involves elastic-plastic deformation and temperature rise in the work-piece and chip, and friction and wear at the tool-chip interface. Therefore, the analysis is based on the following equations:

(i) J_2 flow theory

$$\sigma_{ij}^* = \left\{ \frac{E}{1+\nu} \left(\delta_{ik}\delta_{jl} + \frac{\nu}{1-2\nu} \delta_{ij}\delta_{kl} \right) - \alpha^* \frac{9G^2}{\bar{\sigma}^2(3G+H')} \sigma'_{ij}\sigma'_{ij} \right\} \dot{\epsilon}_{kl}^{\text{mod}} \quad (1)$$

(ii) energy equation

$$\rho C \frac{\partial \theta}{\partial t} = K \left(\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} + \frac{\partial^2 \theta}{\partial z^2} \right) - \rho C \left(\dot{u} \frac{\partial \theta}{\partial x} + \dot{v} \frac{\partial \theta}{\partial y} + \dot{w} \frac{\partial \theta}{\partial z} \right) + Q \quad (2)$$

with boundary conditions

$$\left. \begin{aligned} \theta &= \theta_s \\ K(\partial \theta / \partial n) + h(\theta - \theta_o) + q &= 0 \end{aligned} \right\} \quad (3)$$

Material data should be incorporated with the analysis to obtain realistic solutions, such as

(iii) flow stress characteristics

$$\bar{\sigma} = A(\theta) \left(\frac{\dot{\epsilon}}{1000} \right)^{M(\theta)} \left[\int_{\theta, \bar{\epsilon} \equiv (\dot{\epsilon})} \exp \left\{ -\frac{k\theta}{N(\theta)} \right\} \left(\frac{\dot{\epsilon}}{1000} \right)^{m/N(\theta)} d\bar{\epsilon} \right]^{N(\theta)} \quad (4)$$

(iv) friction characteristics

$$\frac{\tau_t}{\tau_e} = \left[1 - \exp \left\{ - \left(\lambda \frac{\sigma_t}{\tau_e} \right)^{n'} \right\} \right]^{1/n'} \quad (5)$$

(v) wear characteristics

$$\frac{1}{\sigma_t} \cdot \frac{dW}{dL} = C_1 \exp \left(-\frac{C_2}{\theta + 273.15} \right) \quad (6)$$

The finite element discretization using a linear tetrahedral element for the three-dimensional elastic-plastic and thermal analysis is not described here in the interest of space, but the essence is as follows: (1) The overconstraint problem that emerged from using the tetrahedra in the fully plastic range has been solved by extending the Nagtegaal-Rice functional either to a hexahedron which consists of six tetrahedra or to a pentahedron which includes three tetrahedra. (2) On the basis of the equations mentioned above, the effects of elasticity, plasticity, temperature, strain rate, large strain, friction and tool wear have fully been taken into account in the formulation with the iterative convergence method [2] that efficiently yields a steady state of chip flow.

3 Results and Discussion

Figure 1 shows the initial finite element assemblage consisting of 1887 nodes and 7570 tetrahedra. Only the mesh near the tool tip is drawn in the figure though the whole tool has been modelled. The tool was assumed to be stationary and rigid, while the front and bottom sides of workpiece was imposed on the cutting velocity. We have dealt with the three-dimensional turning process of a 18%Mn-5%Cr high manganese steel with a P20 carbide tool. The cutting conditions used are as follows: $V = 80$ m/min, $f = 0.2$ mm/rev, $d = 2$ mm, with a tool geometry of (0,0,6,6,15,15,0.5). Table 1 lists the constants used in the simulation, where the flow stress has a different expression from eq. (4) and the history effects of strain rate and temperature are negligible.

Figure 2 depicts the velocity vector on the workpiece surfaces when a steady state has been achieved. As a result of the calculation, the chip flow is uniquely determined, depending on the presence of the side cutting edge angle and the tool corner, though the chip just after passing through the cutting edges tends to flow in the direction perpendicular to the edges.

Figure 3 shows the distributions of temperature (solid lines) and normal stress (dashed lines) on the tool face. The maximum temperature that reaches up to 900 °C appears at a position twice the feed. The cutting action by the tool corner and the three-dimensional heat conduction result in such a non-uniform distribution. The normal stress markedly increases at the cutting edges, reaching a maximum value of 3 GPa.

Tool wear could be estimated using eq. (6) with the temperature and normal stress thus obtained. Figure 4 shows that the wear rate is strongly affected by temperature since the contour lines are similar to those in Fig. 3. A large wear rate in the vicinity of the cutting edges is due to a high normal stress at the tool tip.

Finally, Fig. 5 is a comparison of the tool face temperature between prediction (solid lines) and experiment (dashed lines). The temperature was measured using an alumina-coated platinum wire of 25 μ m in diameter which was embedded in the tool and created a thermocouple with the chip on the tool face. Since the temperature difference at the center of the depth of cut is about 50 °C at most at both speeds, the prediction of temperature is satisfactory.

4 Concluding Remark

The existing two-dimensional finite element simulation program has successfully been extended to simulate a more generalized three-dimensional metal machining process, such as turning with a single point tool. It follows that the effects of cutting action by the front and side cutting edges and the tool corner upon the cutting mechanism and the tool wear have been clarified. Furthermore, the emphasis was put on using it as a simulation tool in a virtual production environment, having developed powerful pre- and post-processors incorporating computer graphics.

Table 1: Characteristic equation and constants used in the simulation

Flow stress characteristics: $\bar{\sigma} = 3.02\bar{\epsilon}^{0.00714} \{ (45400/\theta + 58.4) + a(1130 - \theta)\bar{\epsilon}^b \}$ MPa where for $\bar{\epsilon} < 0.5$, $a = 0.871$ and $b = 0.8$, and for $\bar{\epsilon} \geq 0.5$, $a = 0.574$ and $b = 0.2$; Young's modulus $E = 206$ GPa; Poisson's ratio $\nu = 0.3$; Friction constants: $\lambda = 1.6$, $n' = 1.0$; Wear constants: $C_1 = 22.45$ MPa $^{-1}$, $C_2 = 21770$ K			
	Thermal conductivity K Wm $^{-1}$ K $^{-1}$	Density ρ kg m $^{-3}$	Specific heat C J kg $^{-1}$ K $^{-1}$
Workpiece (18%Mn steel)	13.8	7950	545
Insert (carbide P20)	66.9	11200	356
Shank (0.55%C steel)	36.0	7750	461

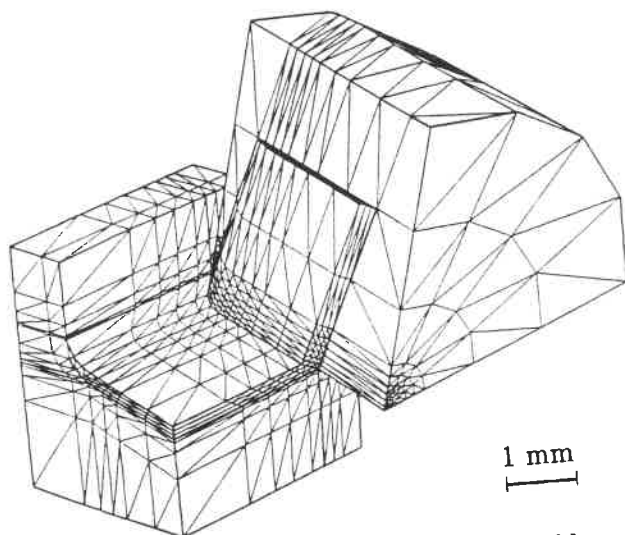


Figure 1: Initial finite element assemblage near tool tip

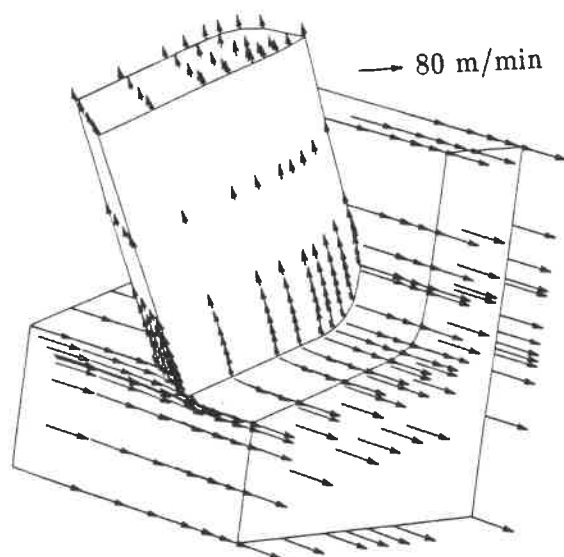


Figure 2: Velocity vector on surfaces

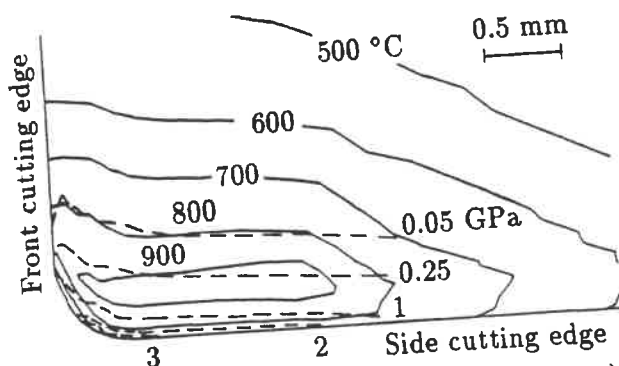


Figure 3: Distributions of temperature (—) and normal stress (---) on tool face

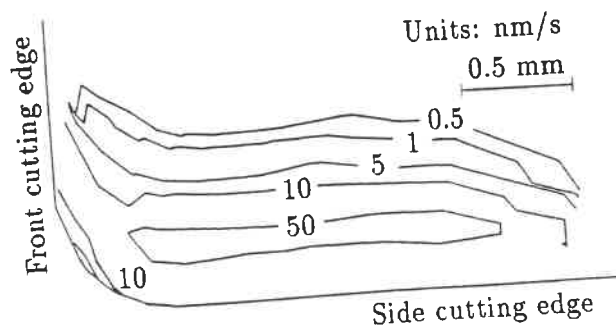


Figure 4: Contour lines of crater wear rate

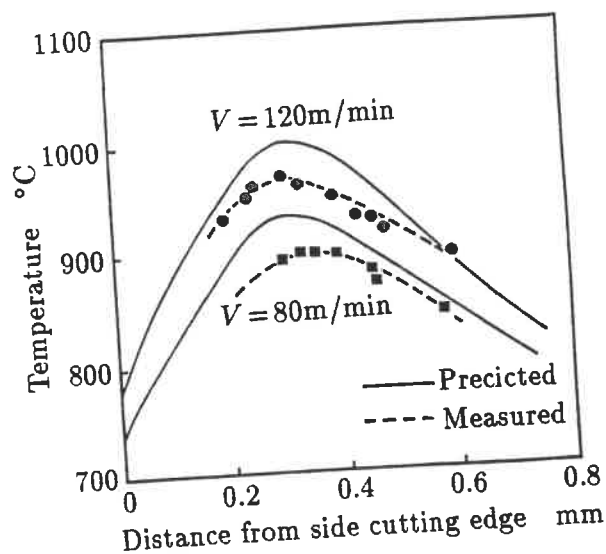


Figure 5: Comparison of tool face temperature in direction of chip flow at center of depth of cut: workpiece, 18%Mn-5%Cr steel; tool, carbide P20 (0,0,6,6,15,15,0,5); $V = 80$ m/min, $f = 0.2$ mm/rev, $d = 2$ mm; dry cutting

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Deformation and Stress Analysis of Gearing

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The gearing belongs to the oldest and most spread mechanisms in mechanical industry. Continuously more demanding requirements are laid upon its service properties. Their solution therefore asks new, high quality materials, new technology of thermo-chemical treatment as the use of new structure types and new computational procedures for gearing structure as it is seen from the system approach to their design. Further effective development of presented problems so requires besides mobilisation of production means to devote increasing attention to the questions of further improvement of gearing design by help of CAD. Besides e.g. shape design of gear wheels, optimal division of partial gear ratio in multistage transmissions etc., we can count also static and dynamic deformation and stress analysis of gears (2-D and 3-D strength gear design - bending analysis and contact carrying capacity).

1. Deformation and Stress Analysis

According to the fact, that question of deformation and stress analysis for many design types of gearing is still not limited by help of standard, it is possible to do its analysis with respect to the basic generally valid elastic relations between basic quantities - displacements, relative elongations, strength and load states. To the mentioned relations we can add those ones going from known Newton's law for dynamic analysis (as a rule in this case it is forced damped nonlinear oscillation movement - system is moving

round certain equilibrium position at action of inside and outside forces).

When solving static deformation and stress analysis the authors aim to mutual comparison of results of solution, reached by help of different procedures and results gained according to standard defined procedures, to assessment of influence of choice of final discretisation of gear region to solution results, to assessment of choice of edge conditions and load conditions on result solution (Fig.1 and 2), to assessment of influence of teeth geometry on result gear strength, to investigation of kind of distribution of load force upon contact line of interactive teeth, on determination of "service" tooth rigidity, on assessment of geometric modification of teeth to result deformations and stress of gearing, to assessment of deformation and stress of gearing at noncomplete contact of sides of interacting teeth.

As the dynamic deformation and stress analysis is concerned, the proper problem formulation, used fysical model and its matematical interpretation is different (it is as a rule devided on problems, which do not take into consideration oil film in contact and problems solved by contact hydrodynamic procedures). In the frame of issued solutions it is studied problems of gear vibrations caused by periodic applied impacts, the different dynamic models of gearing are created with the determination of spectral etc. characteristics, the dynamic forces in gearing are analysed together with the study of dynamic interactions in gearings, e.g. as a consequence of shafts deformation.

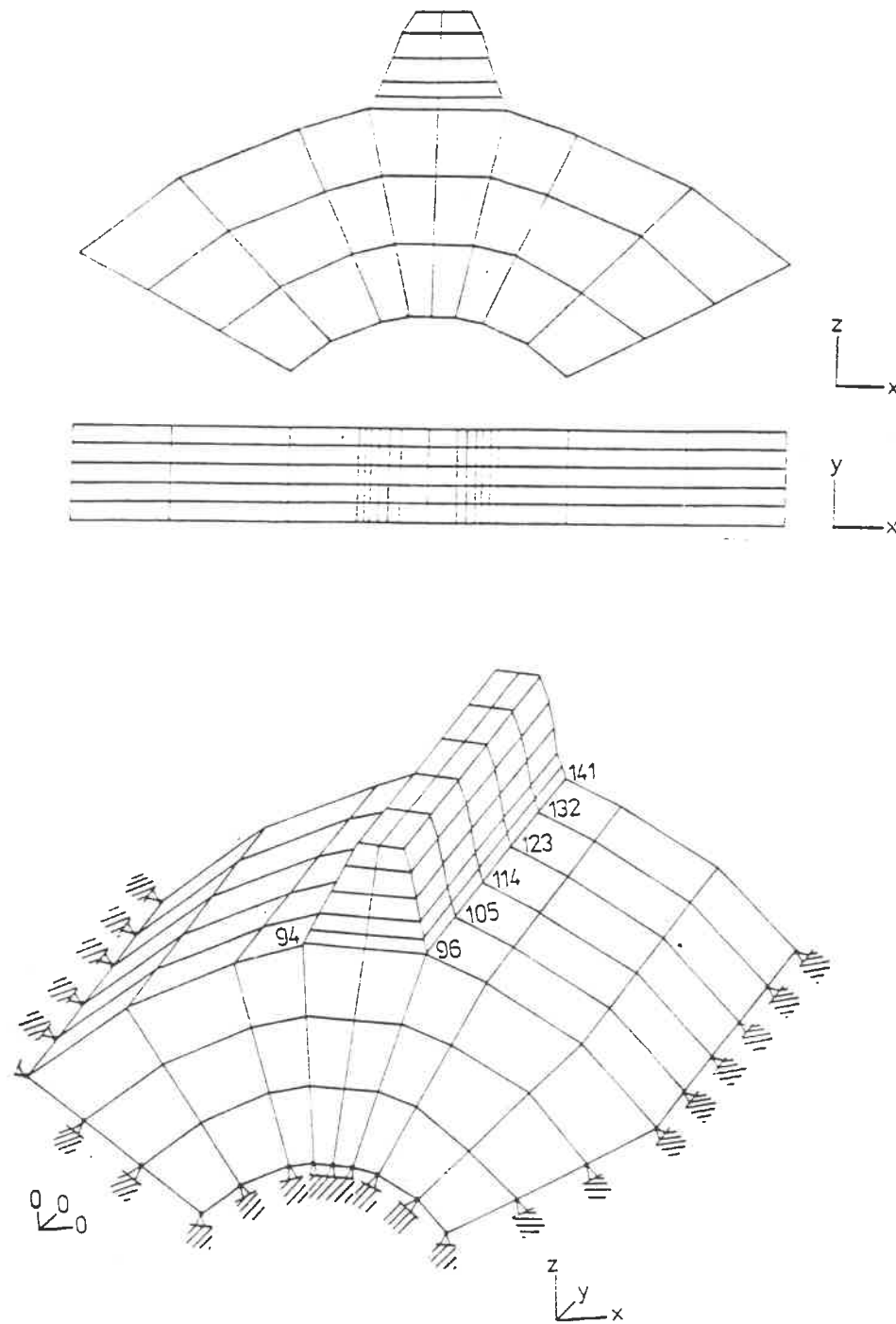


Fig. 1 : The lateral view and view from above to tooth region of spur gearing discretised by 180 space elements (above) together with its axonometric projection and numbering of nodes in root of tooth and choice of edge conditions (down), nonvisible edges considered.

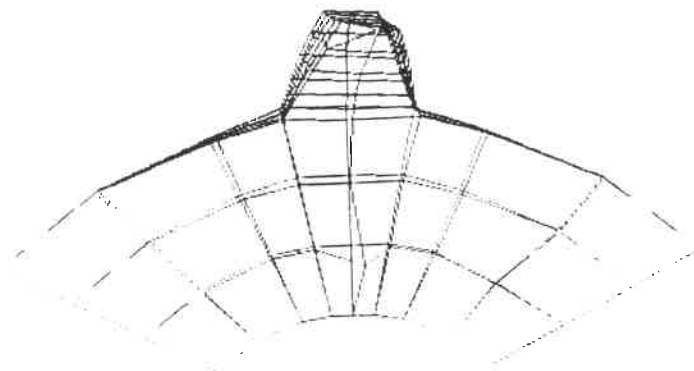


Fig. 2 : The graphic presentation of analysed region of gear wheel deformed by load of unity force F_{xy} (3-D FEA used for solution)

etc. dynamic of contact is studied with the following determination of pressures and their distribution on surface of contact together with limitation of range of local compression, the propagation of surface waves together with the study of thermal and electromechanical effects is done as well as influence of production inaccuracy on dynamic effects in gearing.

2. Conclusion

The set of solved problems from region of deformation and stress analysis of gearing gives the picture of widespread, theoretical demanding requirements and at the same time high actuality of solution of given problems. The right practical application still gained result means and in future will mean the important element of the further development of mechanical parts.

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Molecular Dynamics Simulation of Nanometer-Level Cutting

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We analyze the cutting forces in molecular dynamics (MD) simulation applied to nanometer-level cutting. The result shows that particular features in the force curves correspond to formation of "chip" and "burr".

1. INTRODUCTION

A concept of the so-called micro-machine is a tiny equipment made up of small mechanical parts with active and practical functions. It has great possibilities to change the technology in medical and machine maintenance.

In realizing it, new problems arise to be resolved. First, it is difficult to produce the parts with a desired nanometer-level accuracy, because we do not know a cutting mechanism in the microscopic scale. Second, to reduce the sizes of the parts, we can never neglect either effect of the abrasion or that of the friction [1]. This means that we have to understand microscopic phenomena occurring between two rubbing pieces. There have been previous reports on the molecular dynamics (MD) simulations applied to nanometer-level cuttings of metals [2]. Unfortunately, it seems hard to compare their results with experimental phenomena since they have employed two-dimensional models for the specimen and two-body potential functions for the inter-atomic interactions. We have already applied the MD simulation to phenomena occurring between small tools and Cu specimens [3]. The MD method is an effective technique to demonstrate the behaviors of atoms near surface.

To obtain more realistic result, we adopt the three-dimensional model and the embedded atom method (EAM) function which is suitable for surface, impurities and defects. Our results in the previous paper represented that the cutting produced lumps of Cu atoms such as in the manner of macro-scale cutting. The lumps showed

various shapes and were classified to two types, namely "chip" and "burr". Deformation of the single crystal was observed due to the production of slip and the replacement of atoms. These features of the cutting depend on the process conditions of tool's shapes, cutting velocities and directions. However, mechanisms of the formation of "burr" and "chip" and the deformation are not yet obvious.

In this paper, we study characteristics of cutting force to make clear the mechanism of the formation of "burr" and "chips".

2. MODEL

To describe atomic interactions within specimen, we use the EAM function developed by Oh and Johnson [4]. The interaction between atom i and atom j is represented by a many-body potential depending on the atomic configuration of surroundings of the atoms.

Positions of all the atoms are updated with Verlet's algorithm [5], commonly used in the MD simulation. We do not consider any interactions between the tool and specimen.

Our typical specimen to be cut is small piece of copper single crystal shown in the lower part of Fig.1. It consists of 155 unit cells and each cell has 6 atoms (upper side of Fig.1). During the cutting process, three bottom layers of the specimen are fixed to be held. Kinetic energies of whole atoms are removed by every 5 steps to prevent the specimen from melting. Periodic-boundary condition to $[10\bar{1}]$ direction (y-direction in Fig.1) is applied for an efficient calculation. Cuttings on (111) face

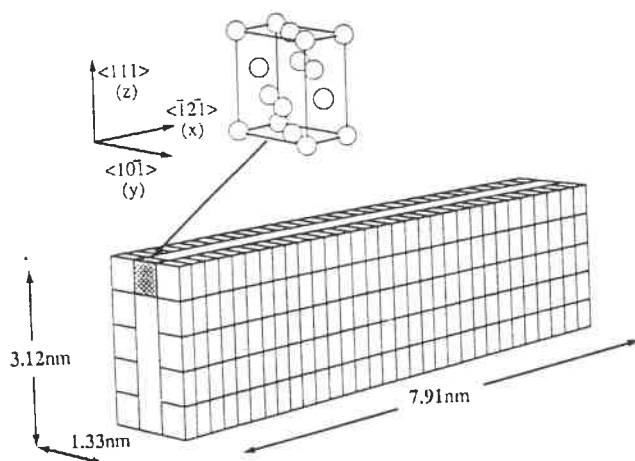


Figure 1. Atomic configuration in Cu unit cell and schematic diagram for specimen to cut before processing.

are carried out along $[\bar{1}2\bar{1}]$ direction (x-direction in Fig.1). We call this model standard one hereafter. We also prepare a tilted cutting face with the angle θ deviating from (111) by $+6^\circ$.

A tool with infinite potential has a round tip with a radius of 0.62 nm and it is set to cut the three top layers of the specimen. Cutting velocities are 50 m/sec of high-speed mode and 5 m/sec of normal-speed mode, which are kept constant during the simulation.

We analyze a reaction force of the tool by dividing it to two components, parallel and perpendicular to the cutting plane. They are called cutting and thrust forces, respectively.

3. RESULTS AND DISCUSSION

Since (111) face is parallel to the slide plane, it is expected that the cutting tool slides the top layers. Fig.2-a shows a result of the standard model at the high-speed mode. The tool is at a half way to cut the specimen. First, we can find a lump in front of the tool. The atomic configuration of the lump shows an ordered structure. The atoms align along to z-direction. There is a disordered structure between the lump and the specimen. These structures are due to the

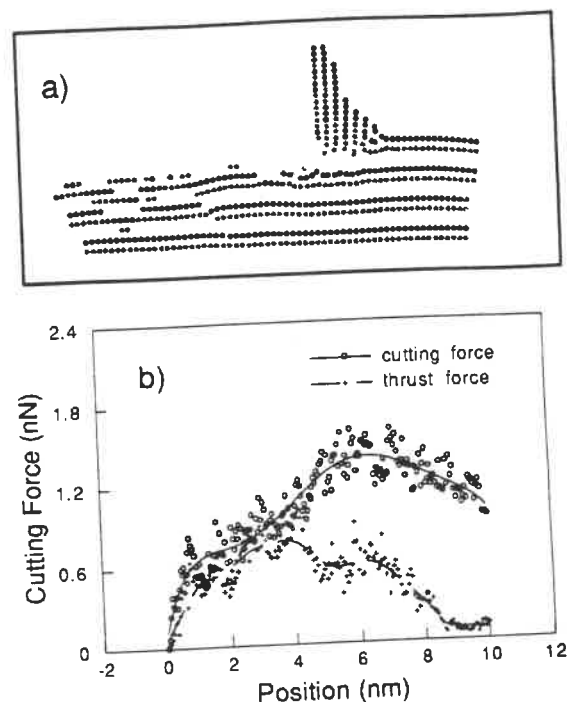


Figure 2. (a) Snapshot of atomic configuration of standard model in the high-speed mode. It is projected onto the x-z plane at a half way of cutting process. Dots in the figure represent individual atoms and a darkness of dots distinguishes atomic layers. (b) The cutting (circle) and thrust force (cross) as functions of the position of the tool. The solid lines are intended only to guide the eye.

way of temperature control. The cutting process transfers an energy from the tool to the specimen; it increases thermal energy, in particular, of the region near the tool edge and it causes a melt. Since we quench the specimen frequently, the melts crystallize again. Hence we do not consider the atomic configuration in the lump so seriously. We can observe no slip along to the cutting direction. These results suggest that the kinetic energy of the tool cause only the melt around the tool.

Second, in Fig.2-a, we can observe a dislocation

located at the center of the specimen. By using animated pictures, it is confirmed that the dislocation is a result of a slip. We also find out a ruin of slip in the figure. A change of the atomic arrangements around the dislocation indicates the slip. There is an intermixing of layers on the left-hand side of the dislocation. (Compare to the configuration of the right-hand side.) We can observe that a group of atoms is forced out to the left-hand side. Because the tool edge pushes the specimen to downward. These phenomena are observed by the animated pictures. They are results of a downward force of the tool.

Fig.2-b shows behaviors of the cutting and thrust forces. Both of them increase rapidly by 1 nm. The origin of the position is an edge of the specimen. The fairly large variations of the forces both at the beginning and at the end of the cutting are due to the finite size of the specimen. Up to the position of 6 nm, the thrust force is nearly constant, while the cutting is immediately increases. Amount of the increase is roughly proportional to a number of atoms in the lump. These results support our former one that most of the energy from the tool is consumed in the lump. Hereafter we call the lump derived from this process as a "chip". We can find out fine structures in the curve of the thrust force in Fig.2-b. We cannot analyze a relationship between the force and the slip since the force curve is noisy. Values of the forces are of an order of 10^{-9} N and they are comparable to the experimental one.

Now, the normal-speed mode drastically changes the formation of lump. Fig.3-a shows a lump is forced out to the right-hand side. It comes from the three top layers sliding to the cutting direction. Hereafter we call this lump a "burr". The forces in Fig.3-b are, in magnitude, one third of the corresponding ones at the high-speed mode. We consider that the cutting force is nearly of Fig.2-b to slide the top layers. A difference in forming the lumps between at the high-speed and at the normal-speed mode is related to an occurrence of the slips. If the cutting velocity is too fast to slide the top layers, only the "chip" forms. It indicates that the formations of "chip" and "burr" could be alternative phenomena. There is, however, a small "chip" in Fig.3-a.

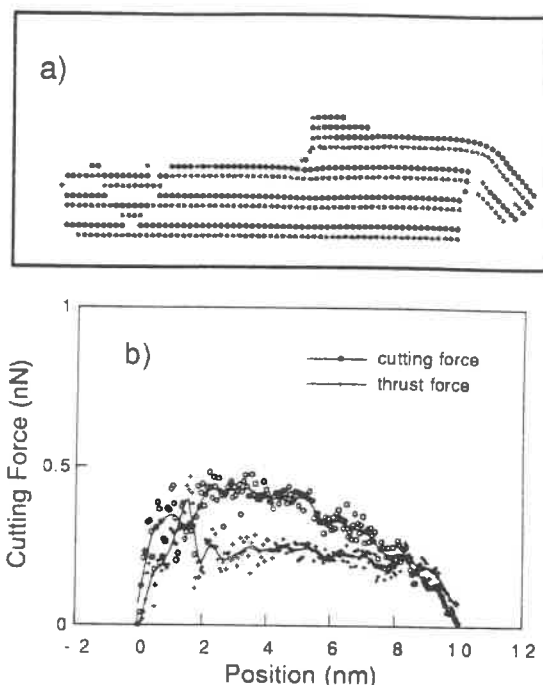


Figure 3. (a) Snapshot of atomic configuration of the standard model in the normal-speed mode. (b) The cutting (circle) and thrust force (cross) as functions of the tool position.

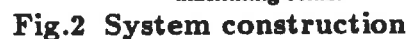
It suggests that there is a velocity at which both of "chip" and "burr" forms.

In Fig.3-a we do not find out any dislocation. In the animated pictures, many slips run, but a number of them is less than in the high-speed mode. The thrust force in Fig.3-b is almost constant and it shows no fine structure except a peak at 1.5 nm. There is a small region showing replacements of atoms at the left-hand side of Fig.3-a. They are similar to the results explained in Fig.2-a. However, region protruded to the left-hand side is smaller than that in Fig.2-a. The replacements and the protrusion occur at the tool position of around 1.5 nm. Our results show that the relation between "burr" and "chip" formations has a dependence on the cutting velocities. An increase of the cutting velocity layers to slide

4. Construction of a sensor information feedback type rapid manufacturing system

fail-safe table [7] are installed on the machining center as critical elements. The rated value of the force sensor used in the experiments is 2,000 N for the main axis direction and 1,000 N for the other directions. The natural frequency of the sensing table is more than 1 kHz and its rigidity is about 500 N/ μm . The output signal from the 6-axis force sensing table is amplified by a strain amplifier and is sent to the real-time controller. The real-time controller calculates various parameters concerning the force vector locus and judges the machining state. The judged information concerning machining state is sent to the "machining condition control task" and the "external NC control task" for adaptive control. Furthermore, the judged machining state is sent to the machining condition determination part and the machining conditions are adjusted to be optimal using rules concerning machining.

In this section, experimental results which correct the machining condition autonomously for full immersion square-end milling are described. In the experiment, the main axis rotational speed and the feed speed are updated for various machining conditions. At the first step, for example, the main axis rotational speed is determined as $3,000 \text{ min}^{-1}$ and the feed speed is 200 mm/min . Chatter is detected from the "force vector locus," as shown in Fig.3(a). A rule concerning machining is applied: "If chatter occurs, the feed speed should be decreased." The feed speed is set as 140 mm/min . However, the chatter is not reduced, as shown in Fig.3(b). Therefore, the feed speed is decreased again, to 100 mm/min . The chatter state is not still reduced, as shown in Fig.3(c). For the subsequent step, the following rule concerning machining was applied: "If chatter occurs, the main axis rotational speed should be increased." The main axis rotational speed was set to $4,000 \text{ min}^{-1}$ while the feed speed remained at 140 mm/min . However, the chatter was still not reduced, as shown in Fig.3(d). Therefore, the following rule was applied: "If the chatter is not reduced, even after the main-axis rotational speed has been increased, the main-axis rotational speed should



be decreased." The main-axis rotational speed was set as $1,000 \text{ min}^{-1}$. Finally, the chatter state was eliminated, as shown in Fig.3(e).

As has been described above, the machining condition was changed to a better one using the judged machining state and the rules concerning machining.

6. Adaptive control using real-time machining condition detection

6.1 Geometrical model generation and cutter path generation for a mechanical part

Fig.4 shows an example of a geometrical model for a mechanical part, especially for an aircraft. The workpiece model includes a pocket at the center and two slots. It is designed using a parametric function, for easy modification of the width of thin plate parts and slots.

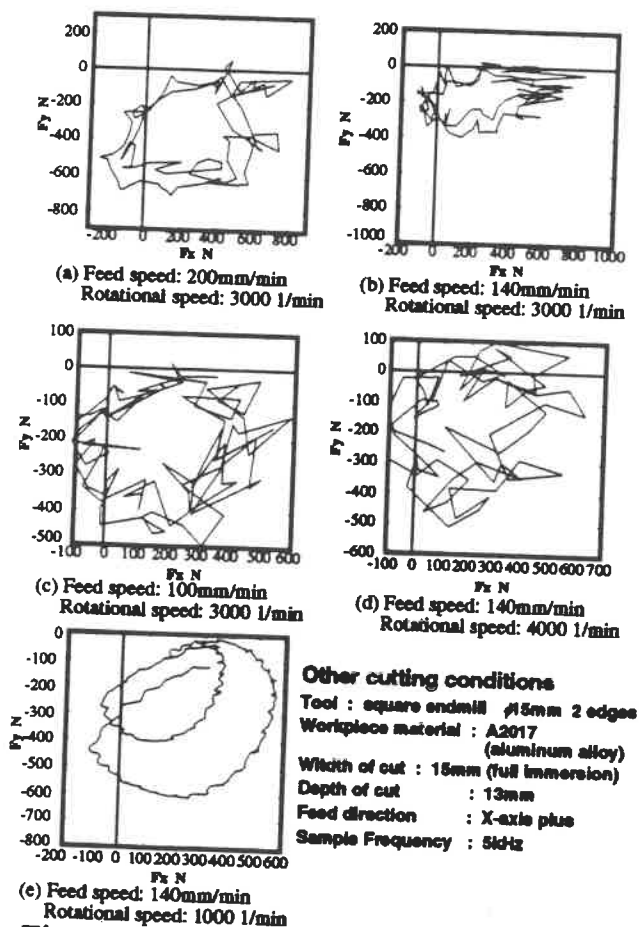


Fig.3 Variance of force vector locus when cutting condition is changed

6.2 Cutting using an autonomous cutting conditions correction function

The mechanical part mentioned above has been machined, modifying the machining conditions autonomously. The machining conditions are changed according to the following rules: [rule 1]: Total machining time is decreased by increasing the rotational speed of the main spindle while keeping the feed speed per tooth constant. [rule 2]: The machining efficiency is increased by increasing the feed speed while maintaining the rotational speed of the main spindle constant. [rule 3]: The cutting force should not exceed the rated value of the tool and the multi-axis force sensor. [rule 4]: If chatter occurs, switch the control mode to the chatter-decreasing control category. This rule is prior to rule 1 and rule 2.

A mechanical part described in the previous section was machined applying the rules mentioned above. Fig.5 shows the changes of cutting force, feed speed, main axis rotational speed and variance of distance in force vector locus when the machining was executed by a square-end mill with a diameter 15mm. The following items became clear from the experiment.

(a) **Machining time** The machining time was reduced by applying a machining condition correction rule which required that the main axis rotational speed and the feed speed be increased while the machining state remains normal. In the case where the mechanical part mentioned above is selected as a workpiece, the 70 minute machining time was reduced to 44 minutes (63%). This result is important because the machining condition is normally determined

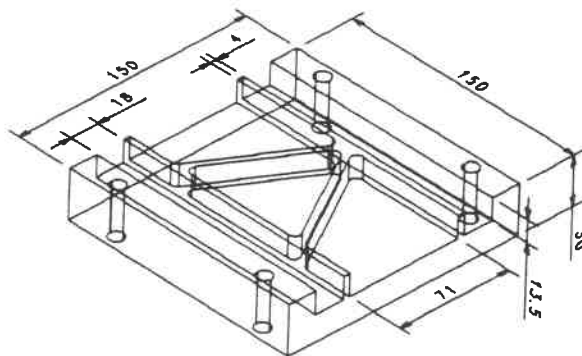


Fig.4 Workpiece model

conservatively when the shape of the workpiece is complicated. The technology greatly reduces the milling time.

(b) Machining precision It was found that the depth of cut became small when the feed speed was controlled too high. It is assumed that the result was caused by the deflection of the tool. On the contrary, the accuracy of the drive surface was fine independent of the feed speed. Consequently, it is required that the feed speed should be constant for finish milling of the part surface, while the rule mentioned above is applicable for the rough cutting and finish milling of the drive surface.

(c) Reduction of the chatter vibration At the machining of the corner of the central pocket (Area 3) in the workpiece, it was impossible to avoid chatter vibration. This result was caused because the machining state was normal until just before the corner, such as in the straight part of the pocket, and therefore the feed speed is controlled to be very high. The result shows the necessity of the prediction of cutting phenomena using the geometrical model of the workpiece and that the main axis rotational speed and feed speed should be controlled according to the prediction.

7. Conclusions

This paper described a sensor information feedback type rapid manufacturing system based on a real-time machining state detection method. The machining conditions were dynamically and successfully changed in real-time in the experiments using the system.

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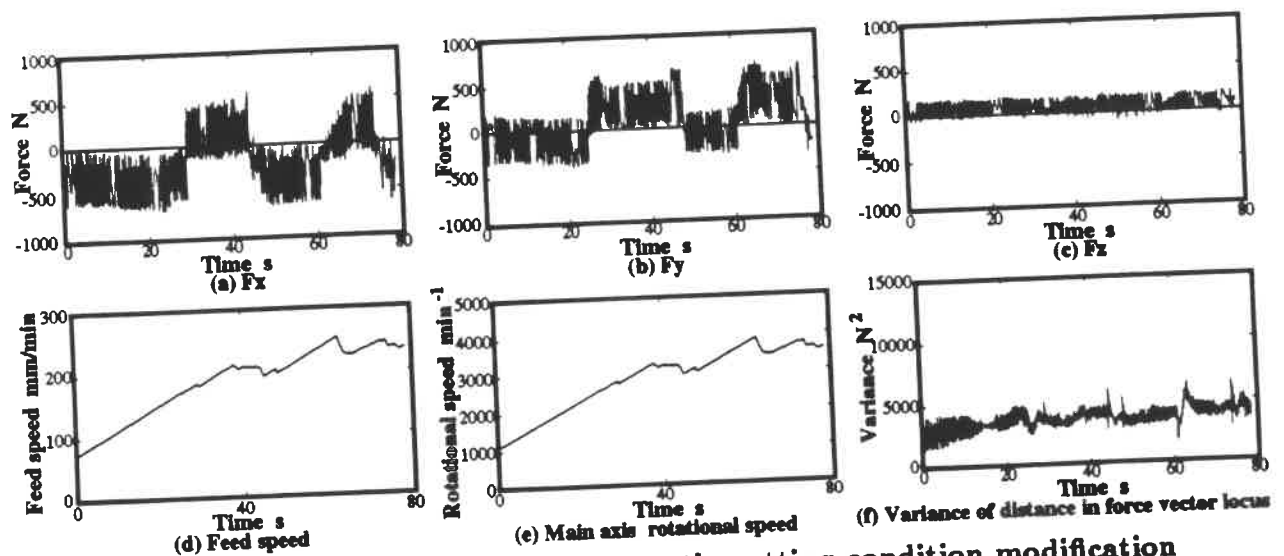


Fig.5 Experimental results of automatic cutting condition modification

Object-oriented rescheduling system in autonomous distributed manufacturing system

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1. INTRODUCTION

With advances of manufacturing technologies in both software and hardware, automation of manufacturing systems in batch production has been much developed and put into practical use. The control structures of the manufacturing systems developed, such as FMS and FMC, are generally hierarchical. The decision making in the manufacturing systems is carried out by a central computer, and controllers of manufacturing equipment only control the individual equipment, such as the machine tools, robots, storages and AGVs (Automated Guided Vehicles) according to the control requirements from the central computer. The hierarchical control structure is suitable for economical and efficient batch productions in steady state, but not necessarily so for very small batch productions with dynamic changes in the volumes and varieties of the products.

The autonomous distributed architectures of the manufacturing systems are therefore proposed aiming at realizing more flexible and adaptable control structures of the manufacturing systems which can cope with the unforeseen changes in the volume and the variety of the products and also the unscheduled disruptions such as breakdown of equipment and interruption by high priority jobs. Some researches have therefore been carried out aimed at developing new architectures of autonomous distributed manufacturing systems[1-6].

The objective of the present research is to develop an autonomous distributed architecture of manufacturing systems. This paper deals with an application of autonomous distributed architecture to rescheduling problems which can modify predetermined production schedules when some unscheduled disruptions occur in the manufacturing systems.

2. MODELING OF AUTONOMOUS DISTRIBUTED RESCHEDULING SYSTEM

Figure 1 shows the concept of the autonomous distributed rescheduling system proposed in the research. The rescheduling system consists of a set of logical autonomous agents representing the physical components, such as machining cells, AGVs, and lots, in the manufacturing systems. The autonomous agents and the physical components are connected by both the monitoring system and the control system.

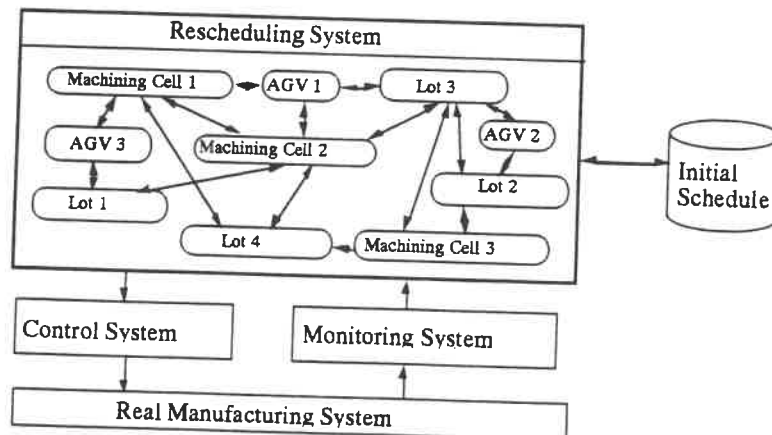


Fig. 1 Autonomous distributed architecture of rescheduling system

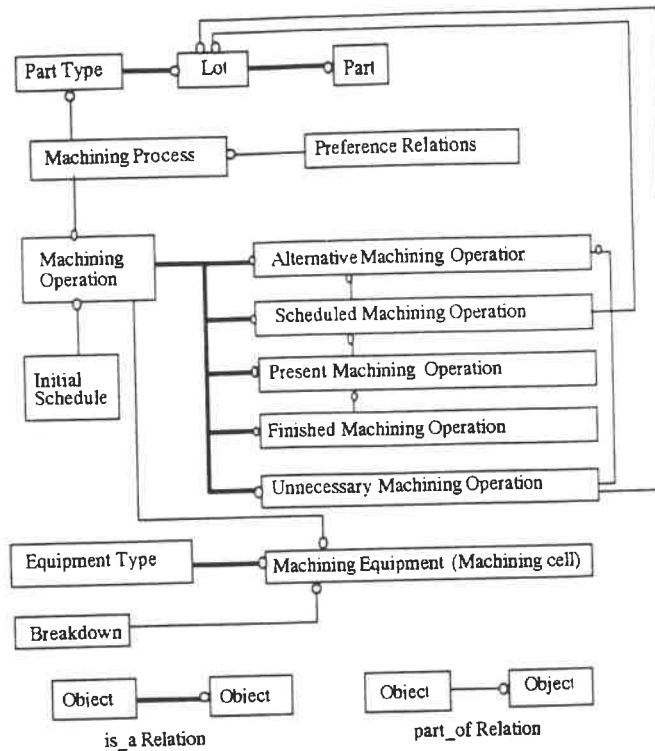


Fig. 2 Objects considered in rescheduling system

The logical agents should have both (1) data representing their static and dynamic status and (2) procedures to make decisions and to exchange the data with the other agents. The knowledge needed in the decision making should also be included in the autonomous agents. It was concluded that the object oriented method is suitable for representation of the dynamic data and the knowledge needed in the integrated process planning and scheduling [7]. Therefore, the object oriented modeling technique is adopted to represent the autonomous agents which carry out the decision making for the rescheduling of the manufacturing systems.

3. RESCHEDULING SYSTEM

3.1 Concept of Rescheduling System

A rescheduling system is discussed here on the basis of the autonomous distributed architecture of the manufacturing systems. The following conditions are assumed for the ease of the rescheduling problem.

- (1) Machining process is considered in the manufacturing systems, but not such processes as the assembly, inspection, and transportation.
- (2) Machining process plans of parts and machining times of individual machining processes are predetermined and have been inputted to the rescheduling system. The process plans of the parts include alternative machining processes.
- (3) An initial production schedule is predetermined and has been inputted to the rescheduling system.

The objects shown in Fig. 2 are needed to carry out the rescheduling of the manufacturing systems. In the figure, the nodes show the objects, and the arcs give the *is_a* relations and the *part_of* relations among the objects. The objects are basically classified into four; those are, the parts, the process plans of the parts, the equipment and the operations.

The objects representing the parts include three classes of objects which are the part types, the lots and the parts. The part type object represents a kind of part to be machined, and the feasible process plans are attached to the part type. The lot objects describe the individual lots of the part types

which are processed in the manufacturing system. The lots are considered as the units processed in the manufacturing systems of batch production.

The manufacturing equipment is described by the objects of the equipment type and the machining equipment.

The process plans of the individual part types are represented by the machining process objects and the preference relations objects. The machining process objects represent individual machining processes, such as turning, milling and boring processes, needed to finish the part types. The preference relation objects give the preference relations among the machining processes, which must be satisfied in order to guarantee the accuracy and quality of the part types.

The machining operations give the relations between the machining processes of the part types and feasible types of machining cells which can carry out the machining processes. The operation time of individual machining process is described in the machining operation objects.

3.2 Rescheduling Activities of Autonomous Agents

The rescheduling system is activated whenever unforeseen disruptive events occur in order to modify the initial schedule to cope with the disruptive events. There are many types of unforeseen events to be considered in the rescheduling system, however, only the event of breakdown of machining cell is taken into consideration here as a typical example of the unforeseen events.

The following rescheduling tasks are carried out when a machining cell is broken down.

- (1) select alternative operations and machining cells for the lots which are being processed by the machining cell broken down,
- (2) determine new schedules of the operations selected in (1), and
- (3) modify the schedules of the other operations, if the schedules determined in (2) disturb the initial schedules of the other operations.

The tasks shown here are carried out through the decision making of the individual agents of the lots and the machining cells and their negotiations under the autonomous distributed architecture proposed here. The procedures of individual agents are summarized in the followings for the case where a machining cell A is broken down.

(a) Lots assigned to the machining cell A at the initial schedule

- select suitable alternative operations for the operations to be carried out by the machining cell A, and insert the operations in the queues of the alternative machining cells.
- request other lots and the machining cells to accept the additional operations in the queues of the alternative machining cells.
- fix the schedule of the alternative operations based on the communication between the machining cells and the lots, if the requests are accepted.
- add the alternative operations to the end of the queues of the alternative machining cells, if the requests are not accepted.

(b) Lots to be machined by the machining cells other than A

- examine whether or not it is possible to accept the requests from the lots of (a).
- modify the schedules of their operations based on the communication between the machining cells and the lots, if they accept the requests.

(c) Machining cells

- examine whether or not it is possible to accept the requests from the lots of (a).
- modify the schedules of their operations based on the communication between the machining cells and the lots.

3.3 CASE STUDY

A prototype of rescheduling system has been implemented in Smalltalk on a workstation. Some case studies have been carried out to verify the capabilities of the autonomous distributed

rescheduling system proposed here. A FMS composed of two machining centers and two turning centers is considered, and six lots are inputted to the FMS in the case studies. Figure 3 (a) shows an example of initial schedules predetermined by a scheduling system [2] which is shown in a form of Gantt chart. It is set that a machining center MC1 is broken down, and the rescheduling system is activated. A schedule modified by the rescheduling system is shown in Fig. 3(b). The figure shows that a part of the operations of MC1 are assigned to the alternative machining cells due to breakdown of MC1.

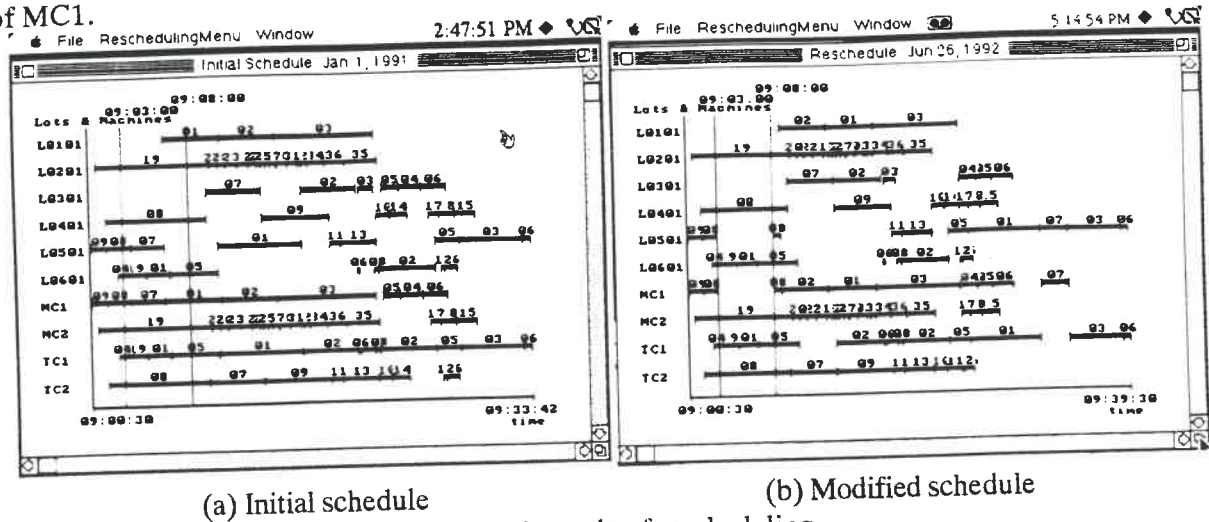


Fig. 3 Example of rescheduling

4. CONCLUSIONS

- (1) A object oriented modeling method is proposed to describe the information processing functions of the autonomous agents of the rescheduling system.
- (2) Objects needed for the rescheduling system are established, which represent the information and the activities of the machining cells, lots, process plans and operations. The relations between the objects are clarified and described by the is_a relations and the part_of relations.
- (3) A prototype autonomous distributed rescheduling system has been implemented in Smalltalk and applied to some rescheduling problems. It was shown that the proposed method can cope with the disruptions effectively.

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Survey and Improvement on the Productivity in Turning Operations Done in Manufacturing Industries by Using NC Program Simulator

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Summary An NC program simulator is developed for evaluating productivity in NC lathe turning. The productivity of 193 NC programs are investigated by comparing the percentage of jog feeding time and rapid feeding time during turning operations. The result of the investigation shows that the jog feeding time and rapid feeding time occupy 82.8% and 10.2%, respectively, of the total operation time on the average of 193 examples. Improvement in the productivity of 193 NC programs are simulated for three cases, and it is verified that the most effective method among the three is the touch and cut method. It can be realized by eliminating an air-cut travel during jog feeding. The total operation time can be reduced by 6.4% on the average using this method. The cutting experiments for estimating tool life proved that the touch and cut method does not reduce the tool lifetime under the experimental cutting condition.

Keywords : NC program, NC lathe, turning, productivity, simulation, air-cut travel

1. Introduction

Once an NC program is made it is never improved in many cases because redundant reprogramming tasks are necessary. Thus, the quality or productivity of the NC program has been overlooked, and the capability of advanced NC machine tools or tool materials has not been performed under the best condition. Of course, automatic programming techniques have been developed in order to reduce the work of the programmer and to minimize the incidence of the programming error./1/ However, APT, which is well known as an automatic programming technique, requires very large and fast computer installations and has not spread widely among the factories of small and medium enterprises. As a result, the quality or productivity of the NC program depends on the ability of the programmer. Our final goal is to develop the NC program diagnosis system for turning which indicates the productivity of NC lathe turning to the programmer.

2. NC program diagnosis system for turning

The NC program diagnosis system presents the programmer with some information about the bottleneck of NC lathe turning. Figure 1 illustrates the construction of the diagnosis system. The NC program simulator has been developed to retrieve machining information about cutting parameters, tool paths, and operating time from an NC program. The simulator can display a tool path chart and an operating time chart on CRT as the

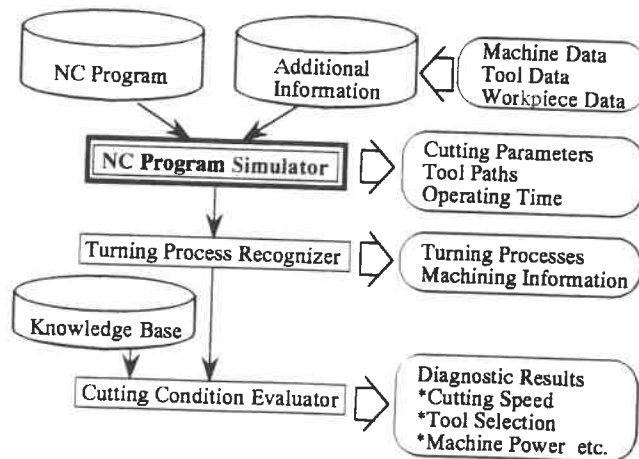


Fig.1 NC program diagnosis system for turning

results of simulation. This information is very useful for determining the productivity of NC lathe turning. The turning process recognizer has been developed to recognize turning processes and machining information involved in NC lathe turning. The turning process is represented by successive tool paths, and each of them plays an important role in the turning process. The recognizer can detect turning processes from successive tool paths and their meanings using syntax analysis. The cutting condition evaluator, which is based on the expert system for turning, shows the programmer the diagnosis results about the cutting condition for each turning processes.

3. Survey on productivity in turning operations

A total of 193 NC programs were collected from 29 manufacturing companies, and their productivities were investigated by comparing the percentage of jog feeding time during turning operation. The 193 NC programs were individually analyzed by the simulator. Figure 2 shows an example of operating time chart displayed on CRT. It is easy to find which operation takes a long time or which operation should be improved. The operating time estimated by the simulator corresponds to the operating time measured on the NC lathe with high accuracy.

The percentages of jog feeding time and rapid feeding time against the total operation time for turning have been plotted in Fig.3 in order to evaluate the productivity of NC lathe turning. The higher percentage of jog feeding time and the lower percentage of rapid feeding time represent higher productivity in the turning operation. The result of the investigation shows that the jog feeding time occupies 82.8% of the total operation time on the average of 193 examples. On the other hand, the rapid feeding time occupies 10.2% of the total operation time on the average. The percentage of jog feeding time increases and the percentage of rapid feeding time decreases with a longer total operation time. The tendency of these time ratios will be a good criterion to estimate the productivity of other NC programs. The NC program should be improved if it has shorter jog feeding time and longer rapid feeding time compared to the average time obtained here.

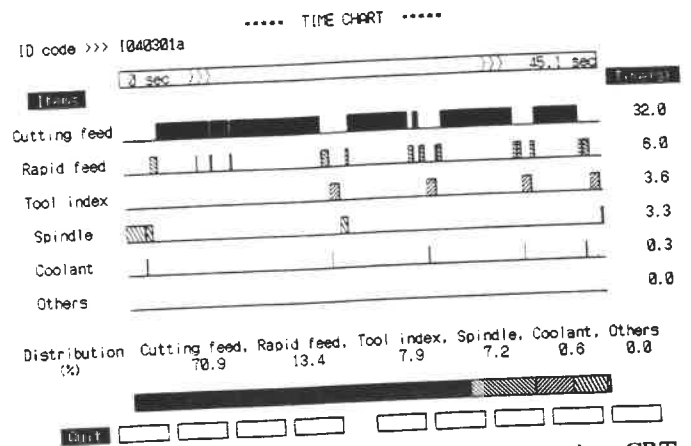


Fig.2 Example of operating time chart displayed on CRT

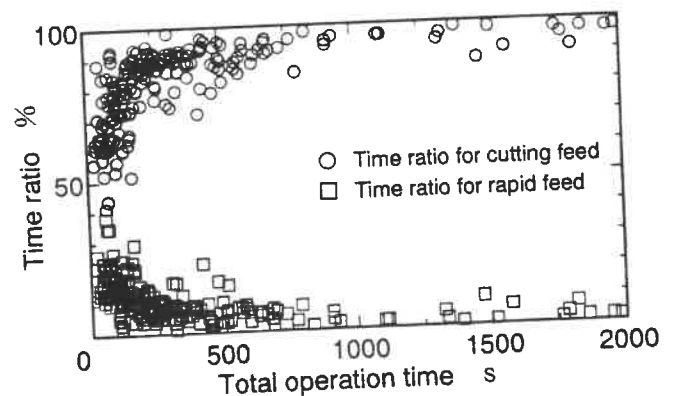


Fig.3 Time ratio for cutting feed and rapid feed occupied total operation time

4. Improvement of productivity in turning operations

Improvement of the productivity of 193 NC programs is simulated for three cases. In the former two cases, rapid feed velocity and rapid feed acceleration, respectively, are assumed to be doubled.

In the latter case, the revised NC programs using the touch and cut method, which eliminates an air-cut travel during jog feeding, are evaluated. Figure 4 shows the results of the simulation.

When rapid feed velocity is assumed to be doubled, the NC lathe turning which has long distance to move with rapid feed can be improved. However the total operation time is reduced by only 1.1% on the average of the 193 examples. When rapid feed acceleration is assumed to be doubled, the NC lathe turning which has many short paths to move with rapid feed is improved, but the total operation time can be reduced by only 1.8% on the average.

In the latter case, the touch and cut method shows that the total operation time is reduced by 6.4% on the average. It achieved the most effective improvement in productivity among the three cases. The reductions of the total operation time for all 193 examples are shown in Fig.5. The NC lathe turning which has many approaches of some tools for rough cutting can be improved in efficiency. The touch and cut method can be easily realized by rewriting the start point of jog feeding which is defined by the NC program.

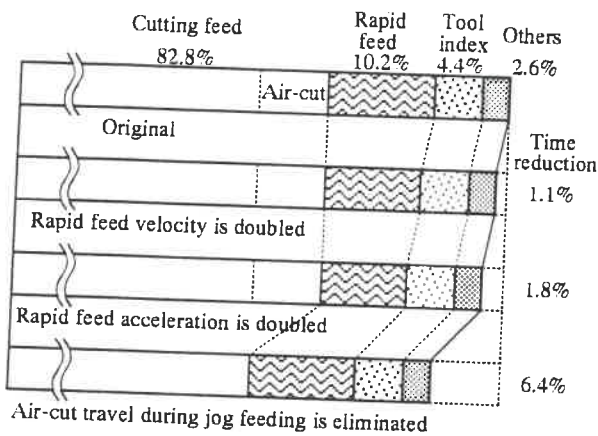


Fig.4 Simulated results of improving productivity

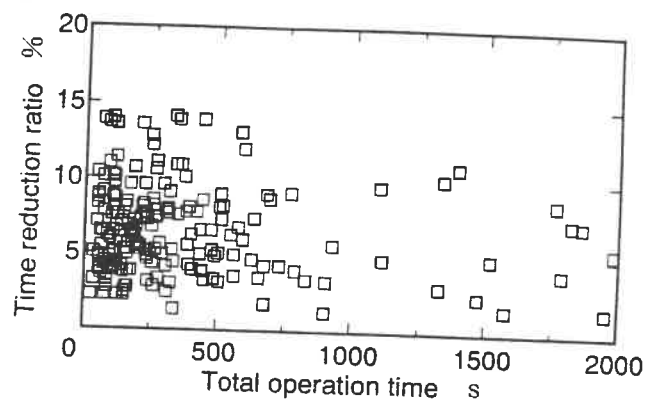


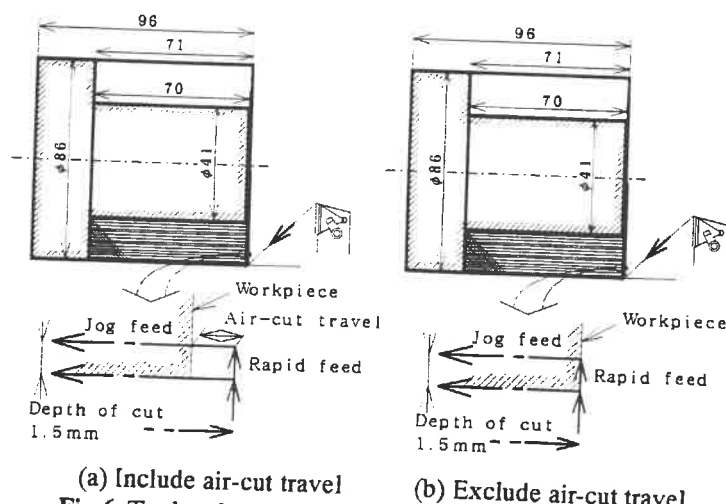
Fig.5 Reduction of operation time when air-cut travel during jog feeding is eliminated

5. Cutting experiment using touch and cut method

5.1 Procedure of cutting experiments

In the touch and cut method, a jog feeding has no air-cut travel for approach of a tool. In this case, tool lifetime may be reduced because a tool comes in contact with a workpiece many times at the start of jog feedings. In order to estimate tool life, the cutting experiments were conducted by the conventional method and by the touch and cut method.

The cutting experiments were performed using a 3.7kW TAKAMAZ TOP-TURN CNC lathe. Rough turning operations



(a) Include air-cut travel (b) Exclude air-cut travel
Fig.6 Tool paths experimented by conventional method and by touch and cut method

were conducted on 86 mm diameter and 96 mm length bars of medium carbon steel (S55C, Hv240). Non coated carbide inserts (P20) were used in the experiments. ISO standard TNGA160408 was clamped in a CTANR2020 tool holder. Five carbide tool tip were used for the conventional method and the proposed method, respectively. Rough turning was conducted under the following cutting conditions; no cutting fluid was used (dry), cutting speed was 200 m/min (constant), depth of cut was 1.5 mm, and feed rate was 0.2 mm/rev. Five workpieces were cut by each tool tip, and a total of cutting length was approximately 5000 m. A comparison of tool paths experimented by the conventional method and by the proposed method is shown in Fig.6. The tool tip experimented by the touch and cut method came in contact with workpieces 75 times during rough turning.

5.2 Results and discussion All of the tool tips had no chipping during experimental cutting. The flank wear of the tool tips was measured to compare tool lifetimes. Figure 7 shows the progress in wear of the tool tips. A hollow circle and a solid circle represent the mean value of the flank wear measured on the five tool tips for the conventional and the proposed method, respectively. The tendency of the wear progress is very closed, and it proved that the touch and cut method does not reduce the tool lifetime under the experimental cutting condition.

As mentioned above, the touch and cut method is easy to realize and is favorable compared to the conventional method. The touch and cut method is concluded to be more effective to improve the productivity of NC lathe turning.

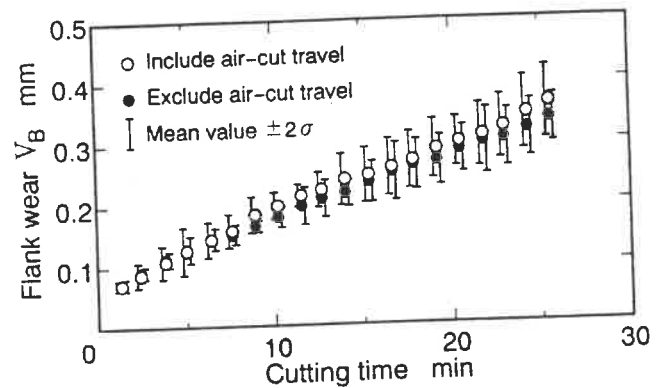


Fig.7 Wear progress of tool tips during experimental cutting

6. Conclusion

An NC program simulator has been developed for evaluating productivity in NC lathe turning. A total of 193 NC programs collected from 29 manufacturing companies were investigated for their productivity by comparing the percentage of jog feeding time and rapid feeding time during turning operations. The result of the investigation shows that the jog feeding time and the rapid feeding time occupy 82.8% and 10.2%, respectively, of the total operation time on the average of 193 examples. It will be a good criterion to estimate the productivity of other NC programs.

Improvement in the productivity of 193 NC programs was simulated for three cases, and the touch and cut method which eliminates an air-cut travel during jog feeding shows that the total operation time can be reduced by 6.4% on the average. It is the most effective method to improve the productivity among three cases. Finally, in order to estimate tool lifetime, the cutting experiments were conducted by the conventional method and by the touch and cut method. The flank wear of the tool tips shows that the touch and cut method does not reduce the tool lifetime under the experimental cutting condition.

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Capacitive Position Transducers; Possibilities in Modern Machining and Manufacturing.

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ABSTRACT

This paper describes the features and design process of capacitive sensors for application in precision manufacturing machines. The basic idea in the design is to realize sensors that perform with high repeatability and a smooth error curve in the primary output signals. Further improvement of the performance can be achieved by using spatial averaging and smart electronics. To enable this approach complete modeling of sensors is required. Three capacitive sensors, realized at the Laboratory for Micro Engineering, are shown.

INTRODUCTION

As the manufacturing processes are automated to a higher degree, the use of computer controlled equipment becomes more important. A very important part of a CNC machine are the position and displacement sensors. More and more capacitive sensors are used in modern machining and manufacturing; mostly for the measurement of small displacements, for positioning but also, for instance, in calipers. This recent development is based on the application of modern electronics, the use of signal processors and computers.

The working principle of a two-plate capacitive sensor is simple and based on the capacitance relation $C = \epsilon A/d$ [1,2]. For easy understanding the lower plate can be seen as the emitter electrode and the upper as the receiver electrode (see figure 1). Many measuring principles are possible by varying one or more of the parameters in the above formula: ϵ = dielectric constant, A = effective overlapping surface and d = plate distance.

The electronic measurement is based on the relation $C = Q/U$ and in practice realized by three basic principles: 1) detecting the current i ($=dQ/dU$) that is induced on the receiver plate by a varying voltage on the emitter electrode, for example in a transformer bridge; 2) detecting the charge/discharge time constant (RC-time) or 3) by detecting the oscillation frequency of a RCL network.

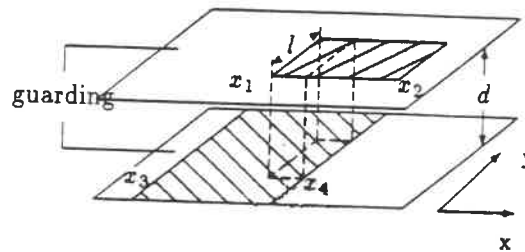


Fig. 1: A two plate capacitor

TYPICAL FEATURES OF THE CAPACITIVE POSITION SENSOR

- A low power consumption allows battery operation.
- Can be made very compact and can even measure directly relative to the object. In practice, down scaling is only limited by a minimum nominal capacitance value of about 0.1 pF (resolution: 10^{-5} pF). The measuring instrument should be mounted directly on a relevant reference as close as possible to the object and should have a minimal Abbe-offset.
- Flexible in design: one basic electronic measuring unit can be used in combination with many types of capacitive sensors. A more complex capacitive sensor can measure many parameters at the same time. For example: primary measurement of X displacement and additionally measurement of the Y, Z, Rx, Ry and Rz displacements; using multiple receiving electrodes.
- A pattern of capacitive electrodes can be designed as a form standard for one or even two dimensions and perform with a very high position repeatability. It must be realized that often

repeatability is more important than absolute accuracy. This can be applied very effectively in step and repeat systems, but also in plotters, printers etc.

- Spatial averaging is an important design tool that can be applied effectively to reduce the influence of production tolerances.

For example: For a measuring scale the length of the ruler, the form standard, in principle only is limited by production facilities but it is also possible to connect several rulers to achieve a larger range. However, the length of the sensor element (receiver plates) can be made relatively long compared to optical rulers, which are limited by the size of the optical source and detector. So, by applying spatial averaging the output error-curve of a capacitive sensor can be very smooth and consequently easy to calibrate by using a curve-fit or a look up table. In practice the Laboratory for Micro Engineering realized measuring accuracies 5 to 100 times better than the production tolerances! It must be realized that fine mechanics are much more expensive than electronics.

When the possibilities for use in manufacturing and measurement equipment of capacitive position sensors (Cap.) with a laserinterferometer (Laser.) and an optical ruler (Optic.) are compared the following can be found:

	Cap.	Laser	Optic.		Cap.	Laser	Optic.
commonly used/available	+-	+	+	length standard/reference	+-	++	+-
environment: T, p, %RH, etc.	+	+-	+	use Abbe principle	+-	+	+-
close to object	++	+-	-	easy to use and to mount	+-	++	+-
space requirement	+	-	-	costs	+	-	+-
integration in mechan. design	++	-	-	absolute accuracy	+-	++	+-
repeatability	++	+	+	Multi dimensional meas.	++	-	-

Note: this table is not universally valid, the criteria depend on the application.

In recent years we have seen that in some precision instruments and manufacturing equipment laserinterferometers are replaced by optical rulers or even by 2 dimensional optical sensors. The authors expect that also capacitive position sensors, specially the more dimensional ones, will be used more and more in the design of precision instruments.

DESIGN AND IMPLEMENTATION OF CAPACITIVE SENSORS

The design of a capacitive sensor needs a mechatronic approach [1]. First, the mechanical and control engineers have to define the required specifications and mechanical dimensions. Then after selecting a physical/electrical/mechanical working principle a lot of modeling is required to optimize the influence of all physical, mechanical and electrical parameters on the complete system. A working physical principle is not enough. The next step is complete modeling to predict the final sensitivity to all mechanical tolerances and environmental conditions is a very difficult multi-disciplinary task.

A practical approach could be the following:

Mechanics + Measurement & Control:

- Mechanical specification of:
 - Input parameter(s): X, Y, Z, Rx, Ry, Rz. (primary inputs as well as parasitic inputs!)
 - Required positioning or position measurement: accuracy, resolution, repeatability, linearity, speed, etc..
 - Overall dimensions: size, weight, moment of inertia, etc.
 - How to mount: Stable, Abbe principle, mechanical reference, etc.
- Measurement & control specifications:
 - Output signals, transfer function, bandwidth. (For high speed control not only position but also velocity and acceleration outputs are needed.)
- Overall constraints:
 - vibrations.
 - material temperature.
 - air temperature, pressure, humidity, etc.
 - electro magnetic interference.

Physics + Electronics (+ Mechanics + Measure & Control):

- Find and/or select a working principle.
- Work out the specifications.
- Split up in different tasks for each discipline; but special attention has to be paid to the multi disciplinary aspects:

- Physics: Electric field modeling: electrode guarding, shielding, gap distance, environmental influences humidity, static electricity: preliminary specifications.
- Electronics: capacitance measuring; reduce influence of parasitic capacitances and EMI; use of micro controllers for signal processing, switching, curve fit and/or look up tables; electronic outputs and interfacing: preliminary specifications.
- Mechanics: design capacitive electrodes as a form standard including guarding, shielding and electrical connectors; design mounting and housing of electrodes inclusive electrical wiring; preliminary tolerances and specifications
- Measurement & Control: required input signals; transfer function: preliminary output and control specifications.
- Multi-disciplinary feedback and complete modeling to optimize all parameters and the final performance.
- Final complete electro mechanical model and specifications.

Finally a test setup including a calibration facility is used to verify the model. Then a design prototype is build and the overall specifications and the final cost estimations are calculated.

EXAMPLES OF SENSOR PRINCIPLES

Modeling the influence of all the mechanical parameters is a very complex task. Once a complete model has been realized it can be used to design other capacitive sensors. As an example, the most important mechanical parameters that have to be considered in the electrode design are given below:

- Electrode pattern: pattern design for optimal performance including shielding and guarding: form/pattern accuracy and stability; flatness; edge quality; roughness, minimum gapwidth.
- Electrical connections: wiring will cause parasitic capacitances so mostly double- or even multi-layer structures are needed. (Capacitive coupling is a contactless alternative.)
- Mounting and alignment of electrode plates: kinematic mounting and the use of symmetry are used as the basic guidelines.

Three position sensors realized will be discussed. Measurements with a Hewlett Packard 5528A laser interferometer were conducted in a temperature and humidity controlled room; all data are based on 3 sigma values. To obtain absolute accuracies look up tables are used. For a more detailed description the reader is referred to the papers mentioned in the references.

Linear displacement sensor made out of flexible single layer printer board [3]. (Figure 2).

Range:	150	mm
Production tolerances:	0.015	mm
Mounting / alignment:	0.2	mm
Absolute accuracy:	0.003	mm
Repeatability:	0.001	mm
Resolution:	0.0007	mm

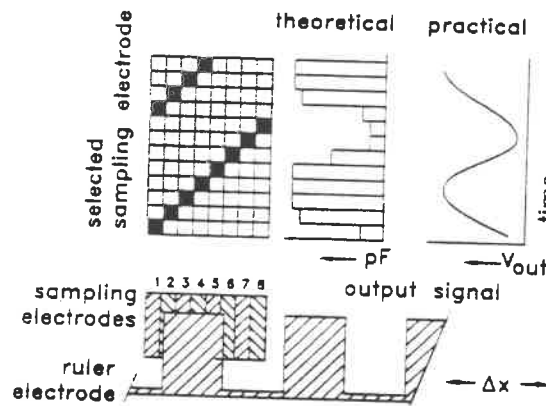


Fig.2: Configuration of the electrodes and generation of the output signal at standstill. The "ruler" electrode is the emitter and the "sampling electrodes" are the receivers.

Linear displacement sensor with a two layer electrode pattern on a glass substrate [4]. (Figure 3).

	coarse:	fine:	
Range:	60	3	mm (20 times)
Production tolerances:	0.01	0.01	mm
Mounting / alignment	0.01	0.01	mm
Absolute accuracy over 10 mm:	0.01	0.00010	mm
Repeatability:	0.05	0.00006	mm
Resolution:	0.03	0.00004	mm

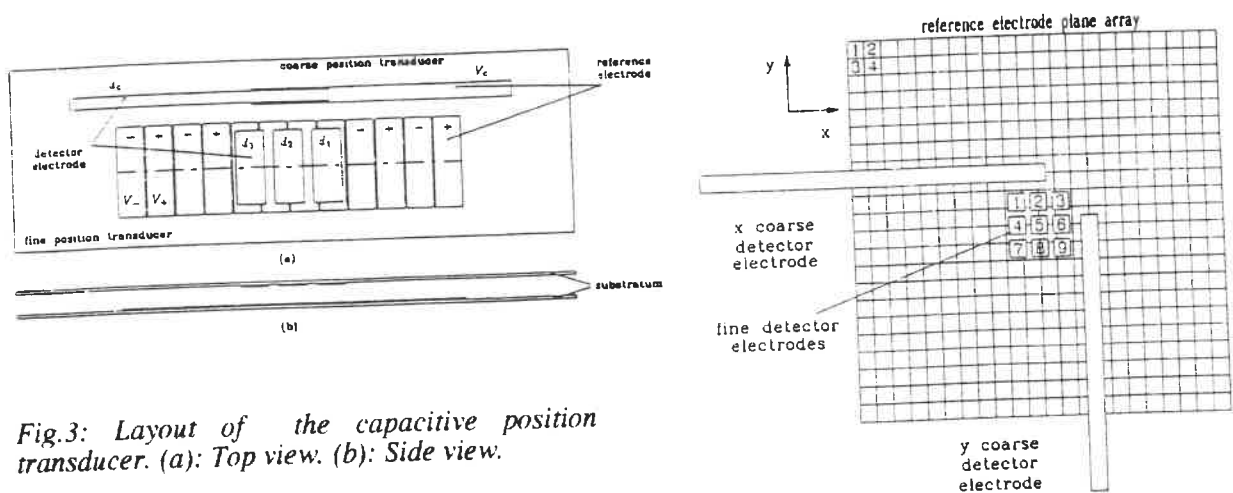


Fig.3: Layout of the capacitive position transducer. (a): Top view. (b): Side view.

X-Y position sensor made out of double layer printed circuit board [5].

A square pattern of 10x14 emitting electrodes is used as a reference or form standard; nine receiving electrodes are used. (Figure 4).

Range X and Y:	60 mm x 85 mm
Production tolerances:	0.2 mm
Mounting / Alignment:	0.05 mm
Flatness:	0.01 mm
Parasitic displacements:	0.01 mm
Absolute accuracy:	0.005 mm
Repeatability:	0.001 mm
Resolution:	0.0005 mm
Step & repeat features:	(10x14x9= 1260 positions in XY)
Repeatability:	0.0001 mm
Local resolution:	0.00005mm

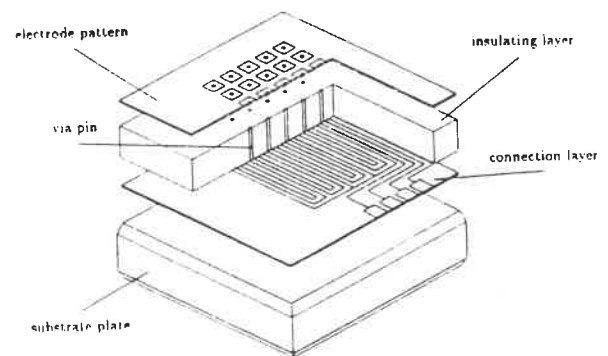


Fig.4: (a) The electrode pattern of a two dimensional position transducer. (b) The multilayer structure of the reference electrode.

CONCLUSIONS

Optimal capacitive sensor design must be integrated in the instrument design. Many advantages and inexpensive solutions are possible but it must be realized this is a multi-disciplinary task. Complete modeling to predict the final output and sensitivity for all mechanical tolerances and other parameters is required. The use of signal processing and look up tables is more effective and less expensive than compensating using fine-mechanics. For absolute capacitive positioning a special calibration facility is needed as part of the production. In this case the calibration of the capacitive form standard is part of the production. For step and repeat applications a very high repeatability can be obtained with limited equipment.

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